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A Survey of Problems of the Thermal Barrier

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In this survey, overall problems of the thermal barrier in connection with pilotless and piloted aircraft are first considered. Individual problems associated with various systems are then presented. Finally, new aspects of structural design and analysis caused by aerodynamic heating are discussed and the associate weight penalties are considered.*

SURROUNDING a moving aircraft is a relatively thin boundary layer of air, in which the still air is accelerated to the velocity of the aircraft. The heating associated with the frictional effects in the boundary layer is rapidly dissipated in subsonic flight. In supersonic flight, the effects become noticeable but are generally of little importance below Mach 2 because of the fact that flights usually are made in the stratosphere, where the ambient temperature is —67° F under standard conditions. Beyond Mach 2, the effects become progressively more severe.

High temperatures are not entirely a new problem to aircraft and have affected the airframe near engines and thermal de-icing equipment. At speeds beyond 1500 mph, however, the elevated temperatures caused by aerodynamic heating become an overall design problem. Thus, for example, aluminium melts at Mach 5, and steel melts at Mach 6.

With regard to the ultimate effects of the thermal barrier on aircraft performance, some contend that a suitably designed aircraft or missile may fly very fast at extreme altitudes, where the density of air is practically zero, and then proceed at reduced speed while passing through lower altitudes. During the short periods of flight in the denser regions of the atmosphere, even very high stagnation temperatures may not do much harm if the structure has a great deal of heat capacity, if it is protected by a layer of insulating material, or if it is cooled.

Others have come to more pessimistic conclusions, contending that large temperature increases and heating rates render any method of preventing the primary structure from heating up for more than a short period of time, at best, only palliative. Studies have shown that the weight penalties involved in insulating or cooling the structure are tremendous. Considerations of the use of insulation, heat storage, cooling, and other methods indicate that sooner or later the aircraft engineer will have to face the task of designing primary structures to carry loads while hot.

(1) OVERALL EFFECTS

(A) PILOTLESS AIRCRAFT[1]

A vehicle travelling in a ballistic trajectory at very high speeds is in the relatively high-density air where heat transfer and drag are important for only a short time. The real problem arises in the re-entry phase of the trajectory, where, for a brief period, the vehicle is subjected to enormous values of heat transfer, and it becomes of vital importance whether the boundary layer is laminar or turbulent. For the re-entry, a relatively heavy low-drag vehicle which decelerated very little was assumed in an example. The maximum laminar heat-transfer rate is over 2,000,000 BTU per hr per sq ft, which is of the order of 50 times that ordinarily encountered in modern boilers, whereas, in the case of turbulent boundary layers, this figure is of the order of 1000 times.

Several possibilities of relieving aerodynamic heating present themselves. Reverse thrust, to decelerate the vehicle before it enters high-density air, would be usable, if sufficient fuel could be carried. However, an increase in initial weight of at least one order of magnitude would not be unlikely. The vehicle may also be decelerated at high altitudes by the use of a large drag-to-weight ratio. This is the mechanism by which small meteorites (0·04 in. in radius) reach the earth's surface, even though they have much higher entry velocities than those considered for man-made vehicles. Application of the same principle to a large vehicle would require that the drag be very large at high altitudes, even though the air densities are extremely small, so that the structure required is likely to be large and heavy.

For some flight conditions, the structure probably can be protected to some extent by means of an insulating material applied to the exterior of the vehicle. However, there are no insulating materials known which can stand up under the high temperatures, thermal stresses, and aerodynamic forces encountered at the higher Mach numbers.

Another possible approach is to cool the surface by some means, such as transpiration cooling, where a fluid, possibly water, is forced through the surface. Although the use of a transpiration cooling system or, for that matter, any cooling system, involves considerable design complication, it is a promising approach.

As a further possibility, the structure of the vehicle may be used to absorb the heat, and parts of the vehicle may be allowed to melt, if necessary. This process is one which occurs in nature for very large meteorites, in which aerodynamic heat-transfer rates are high for a short time, but only a small part of the heat can be conducted to the interior, so that melting occurs at the surface, while the interior remains relatively cool.

Model tests:—To explore the last possibility, an investigation was undertaken to melt models by aerodynamic heating in a hypersonic wind tunnel at Mach 6·9, using Wood's metal, which melts at 158° F. For these tests, the free-stream temperature was about — 350° F, while the stagnation temperature was about 700° F. The configurations used included a solid hemispherical-nosed cylinder, a solid cone-cylinder, a thin-walled cone, and a thick-walled cone. All the cone models had a 20-deg. apex angle.

The hemispherical-nosed model, as well as the cone-cylinder, was observed to start melting almost immediately with the start of the airflow in the nozzle

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of the wind tunnel. On the hemispherical-nose model, melting started at the centre of the nose and rapidly spread to cover the entire hemisphere during the first 10 sec, after which the nose shape remained constant for the rest of the run. The stable shape was considerably blunter than the original hemisphere. On the conical-nosed models, melting started on the tip, producing an approximately spherical blunted nose, which grew in radius as it moved back on the cone.

In order to obtain conditions in which melting would occur on some part of the model other than the blunted nose, a thin-walled conical-nosed model was tested. In contrast with the uniform melting observed on the solid models, the entire thin wall failed shortly after the melting of the outside of the wall had started. This indicates that the inner surface of the wall reached the melting point at approximately the same time as the outer surface.

Preliminary tests have shown that the melting is an orderly and reproducible process. Melting occurred only on the nose of the solid bodies in the tunnel, because the heat-transfer rates were sufficiently low over the remainder of the bodies, so that much of the heat could be conducted to the interior. In flight, where the heat-transfer rates can be very high, melting might also occur at parts of the body other than the nose, regardless of the skin thickness. In tunnel tests, melting did not appreciably alter heat-transfer rates.

(B) Piloted Aircraft[2]

There are two basic ways of attacking the aerodynamic heating problem. The first approach is to design the aircraft to endure the high temperatures. This might be possible at low Mach numbers with some structural materials and with simple equipment. It is the simplest solution from the aircraft designer's point of view, because it avoids the necessity for cooling systems except for the pilot, but it appears at the present time to present almost insurmountable problems of equipment development at even moderately high Mach numbers.

The second approach is to keep the aircraft cool, so that conventional designs can be used. Even here, the aircraft designer is faced with two choices. First is the insulation of the aircraft and equipment, with no provision for cooling. The success of this method depends on a low rate of heat rise and a short time of flight, to keep the equipment below its limiting temperature. The second approach is to provide some form of cooling system. Some of those which have been used in the past have been examined to determine their suitability for use in high-speed, high-altitude aircraft. These cooling systems fall into three general classes, i.e., air-cycle systems, water-evaporation cooling systems, and fuel cooling. The latter is possible, of course, only in chemically fuelled aircraft.

Hypothetical aircraft:—In view of the performance which might be expected from cooling systems, consider the choices facing the designer of a hypothetical high-speed aircraft. This aircraft is assumed to have a gross weight of 25,000 lb and a fuel weight of 12,500 lb. Three wing loadings of 100, 330, and 1000 psf, as shown in Fig. 1, have been assumed. It will be noted that the drag curves and the fuel-consumption curves are identical. This is because a fuel consumption of 3 lb per lb thrust per hr was assumed for all Mach numbers. This is about the present fuel consumption of jet engines with afterburners at low supersonic Mach numbers, and theoretical considerations indicate that, if efficient air intakes can be developed, fuel consumptions of this order are still possible at high Mach numbers. At Mach 6 this fuel consumption would correspond to 0·26 lb/hp-hr, which is not impossible, considering the high-compression ratio associated with this speed.

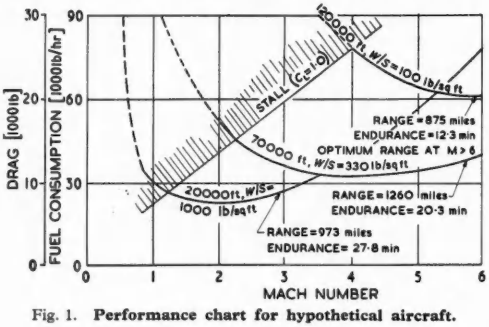
DRAG [1000 lb]
FUEL CONSUMPTION [1000 lb/hr]
MACH NUMBER
STALL (C_L = 1·0)
120000 ft, W/S = 100 lb/sq ft
70000 ft, W/S = 330 lb/sq ft
20000 ft, W/S = 1000 lb/sq ft
RANGE = 875 miles
ENDURANCE = 12·3 min
OPTIMUM RANGE AT M > 6
RANGE = 1260 miles
ENDURANCE = 20·3 min
RANGE = 973 miles
ENDURANCE = 27·8 min

Fig. 1. **Performance chart for hypothetical aircraft.**

The aircraft with a wing loading of 1000 psf at 20,000 ft would have a stalling speed somewhat above Mach 1. Its minimum drag and fuel consumption occur at about Mach 2, and its maximum range accurs at about Mach 3. At 70,000 ft the aircraft with a wing loading of 330 psf would have a stalling speed above Mach 2, minimum drag and fuel consumption at Mach 4·5, and maximum range at a speed of Mach 5·6. Note that this aircraft would have a range of about 1260 miles, which it would cover in 20 min. The aircraft flying at 120,000 ft with a wing loading of 100 psi would have a stalling speed of Mach 4, minimum drag at Mach 5·8, and maximum range at some higher Mach number. At Mach 6 the range would be 875 miles for a flight time of 12 min.

(2) EFFECTS ON SYSTEMS

(A) Turbojet Problems[3]

Although this discussion is confined to turbojet engines, by proper modification for the unique circumstances concerned the data can be applied to other forms of air-breathing propulsion systems. It may be assumed that, at present, man-carrying and some unmanned supersonic vehicles will be turbojet-powered, for reasons of state of development, economy, and missions to be performed.

Compressor problems:—Having taken air on board the engine, it must now be compressed, and this is where the temperature problem becomes acute, as shown in Fig. 2. For instance, a sea-level static 9:1 compressor operating at Mach 1 above 35,000 ft will receive inlet air with a total temperature of about 60° F and discharge the air at about 500° F. However, the same engine operating at Mach 2·8 receives air at about 650° F and discharges it at about 1100° F.

Considering the effect of temperature rise on metal strengths, it is obvious that a much more limited choice of materials is available. Furthermore, the

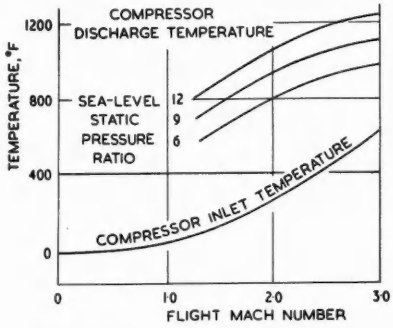

Fig. 2. **Temperature rise through compressor at various flight Mach numbers for compressors of several sea-level static-design-pressure ratios operating above 35,000 ft on a hot day.**

For reference purposes, standard day inlet temperatures are given in the lower curve.

vibratory stresses caused by blade-flutter problems are likely to complicate the design problem still more. Considering that a high-performance compressor may have 1000 blades, exclusive of stators, and that the failure of one blade can completely wreck the engine, the compressor appears as an old problem which has become more critical and difficult of solution under high-temperature conditions.

In addition, in spite of thermal expansion and creep, the designer must provide for the small clearances between blade tips and the casing which make for high compressor efficiency. A further problem arises in providing satisfactory seals to prevent hot-air leakage to the interior of the engine. Thus, the increase of problem areas in future compressor designs cannot be over-emphasised.

Turbine problems:—The turbine is the most critical engine component with respect to temperature. The maximum operating temperature which the turbine can withstand automatically acts as a limit to the amount of energy in the form of fuel which can be added upstream of the turbine. At Mach 1, a turbine-inlet temperature increase from 1600 to 1800° F permits an increase of approximately 22% in fuel addition per pound of air, while at Mach 2·5 the same 200° F increase in turbine-inlet temperature permits a 44% increase.

In the case of an engine designed for relatively low speeds, the difference between compressor-discharge temperature and turbine-inlet temperature allows sufficient differential to add enough fuel to provide compressor power and a significant amount of useful thrust for propulsion. As much as two thirds of the energy represented by this temperature differential is required to power the compressor.

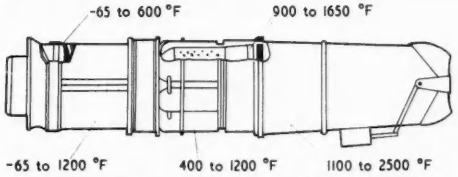

Fig. 3. **Range of temperatures to which turbojet engine components may be subjected.**

Now, consider the high Mach number engine operating at the same turbine-inlet temperature but at a much higher compressor-discharge temperature. The differential, representing energy available for the compressor and thrust, has been reduced considerably. Power requirements, to make up compressor and turbine losses, although probably reduced, will still consume a major portion of the available energy and can conceivably be of such magnitude as to allow no energy for thrust requirements. Therefore, excluding the use of afterburners and other thrust-augmentation devices, it is readily seen that, in order to provide the same compressor discharge-turbine inlet-temperature differential as was provided by the low-speed engine, increased emphasis must be applied to allowing higher maximum turbine-inlet temperatures.

To sum up the overall engine problem and to provide a temperature frame of reference for the engine designer as well as the airframe and accessory designer, practical temperature ranges to be expected are shown in Fig. 3.

It is concluded that compressor designers must now provide inlet-stage blades which might run at temperatures which previously were considered only for last-stage blades. Last-stage blades must operate under conditions as severe as the turbines endured in early engines. As allowable turbine-inlet temperatures are raised, combustor liners and tubes must resist the severe temperature extremes now applied to turbines. The turbine itself, long the real hot-bed of thermal difficulties, forces materials and design engineers into areas of effort where arithmetic performance progress requires geometric effort expenditures. Downstream of the turbine, the constant quest for higher performance and lighter weight sustains the challenge to the designer of aft sections, afterburners, and propulsion nozzles.

(B) Propulsion Installation Systems[1]

Fuel systems:—It is obvious that heat will be conveyed to the stored fuel of a supersonic aircraft, with fuel stored in either the wing or fuselage. With hydrocarbon fuels, certain losses will be encountered unless pressurisation is used. For example, a hypothetical Mach 2 bomber, in sustained flight at 40,000 to 50,000 ft, would lose about 18% of its fuel due to vaporisation. This fuel is assumed to be initially at 100° F, with no tank pressurisation. The addition of $\frac{1}{16}$-in. cork around the tanks would reduce this loss to 8%, again with no pressurisation. However, with insulation and 4 to 5 psi pressurisation, the fuel loss is negligible. Other configurations than that investigated might not be so favourable, and higher fuel-tank pressures might be necessary.

While insulation is important, it could penalise the mission of the aircraft by taking up space which could be utilised for fuel.

The large quantity of fuel aboard supersonic aircraft is regarded by some engineers as a panacea for their supersonic problems, on the grounds that the fuel system is considered an ideal heat sink. However, the designer of fuel systems has more than his share of problems under conditions of supersonic flight, without letting anyone else add heat to the fuel. It must, in some instances, be so used, but all such heat additions must be carefully integrated into the design of the aircraft.

Lubrication systems:—Lubrication systems must be predicated upon some external mechanism of cooling the oil, in addition to operation at the extremely high altitudes of which supersonic aircraft are capable. Oils capable of sustaining high temperatures should have properties of non-inflammability and retention of lubricity. An ideal oil for modest supersonic speeds should be operational in the temperature range from about — 65° to 800° F.

The development of high-temperature oils, however, is not the total solution to the problem, since it is necessary that bearings or other devices which are lubricated operate at very high temperatures. The possibility of designing main bearings to have their own independent lubricating system, and hence eliminating pumps, has merit.

Air-induction systems:—Of particular significance in the installation of a supersonic air-breathing engine is the problem of getting air delivered efficiently to the engine. Supersonic operation must ensure that the shock waves generated at the inlet be properly positioned, in order to obtain high ram recovery, low drag, and even pressure and velocity distribution across the compressor face (or burner inlet for ramjets). The inlet may be of great help in obtaining power for supersonic flight; for instance, at Mach 3, the compression across the inlet shock wave can be about 18 to 1.

As speeds increase, perfection of the air-induction system over the speed range of the vehicle becomes imperative. In the field of variable-area inlets, the need for data which can be developed into practical installations for aircraft, engines, and controls is of the greatest importance.

Cooling systems:—The acuteness of the problem of cooling the engine and its associated systems stems from the lack of suitable non-critical materials which will withstand the high temperatures encountered in high-Mach aircraft. Not only must the power-plant installation itself be cooled, to obtain satisfactory per-

formance of the engine and accessories, but also the structure and aircraft skin surrounding the power plant must be kept within temperature limits.

In addition to aerodynamic heating, the cooling problem of a supersonic aircraft power-plant installation is aggravated by the following considerations:—The larger engines to be used in a supersonic aircraft and missiles have, in many cases, greater compression ratios and hence increased temperatures of the engine case over the last few stages of the compressor. The use of cooled turbine blades and afterburners (with the temperature of the latter running as high as 3500° F) will result in much higher temperatures for the "hot" parts of the engine.

Conclusions:—The power plant and its installation systems, with increasing aircraft speed, account for increasingly larger percentages of the aircraft weight. The fuel system has been assessed as being particularly critical. Supersonic speed adds heat to the fuel system in such quantities that it gives the fuel-system designer many problems; the fuel system is not the ideal heat sink which, at first, it might appear to be. Other power-plant installation-system problems demand early consideration in aircraft design, for compromise in the solution of these problems, owing to their high percentage effect on the range of the aircraft, may well result in compromise of the total mission.

(To be concluded)

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RUSSIA

Electric Slag Welding of Boiler Drums

By I. D. Davydenko. (From *Vestnik Mashinostroyenia*, No. 12, December 1954, pp. 70-73, 3 illustrations.)

The electric slag process of welding is here performed in a special metallic mould, the walls of which are formed by the two vertical edges of the butt joint to be welded and two moving copper slides, which bridge the clearance between the welded edges, both above and below, as shown in Fig. 1. At the beginning of the process, the bottom of this mould is formed by an auxiliary metal plate, tack-welded underneath the butt joint, but, as the process continues, the solidifying metal at the bottom of the welding bath takes over this function.

For the formation of a welding bath, an arc is struck between the electrode wire and the bottom of the mould inside a layer of flux previously heaped in the mould. After some time, the welding bath is filled with molten metal and slag, which, rising in the course of the filling up of the bath with molten metal, shorts the arc when the electrode wire becomes submerged in the slag. At this instant the arc process is transformed into an arcless or electric slag process, in which the electrode wire submerged in the slag is fused by the passing current. From the end of the wire a jet of liquid metal emerges with a speed equal to that with which the wire is fed into the bath, and the current flows through this jet of metal. One part of the welding current branches off in all directions from the wire, and the jet of liquid metal then heats the fused slag to a temperature exceeding the melting point of the metal. The walls of the joint are gradually fused by the rising liquid slag, and the molten parent metal slowly flows down from the walls of the joint and mixes with the metal of the wire. As a result of the mixing of the parent and wire metal, the metal of the welding seam is formed, and this has a barrel-shaped cross-section, which, in an etched condition, shows a uniform appearance without any separate layers.

Crystallisation of the seam metal begins at the walls of the joint and at the bottom of the welding bath at the points of greatest heat removal. In the centre of the seam the metal cools most slowly and for this reason the metal bath has its greatest width at this point. In the preparation of the component for welding, the edges of the butt joint are cut square to the surface of the plate and the two plates to be welded together are joined by clamps which permit displacement of the copper slides.

The gap between the joint edges, subsequently filled with metal, is between about $\frac{13}{16}$ and $1\frac{1}{8}$ in., depending on the thickness of the plate. An automatic welding head, with slides and a flux hopper, is arranged to move along one of the edges of the joint. The seam is welded throughout its thickness in a single pass, and the slides help to form the surface of the seam.

It is claimed that this new method of automatic welding is particularly economical for thick-walled pressure vessels, especially insofar as reduction of power consumption and flux is concerned.

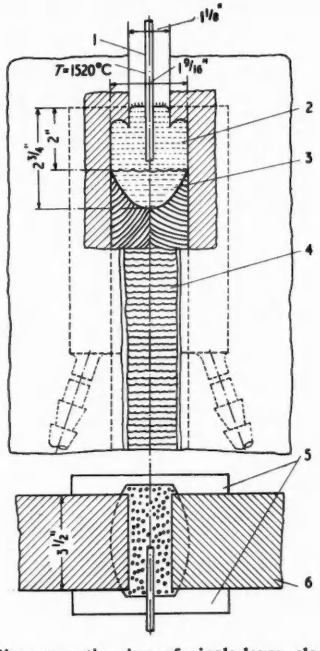

Fig. 1. **Diagrammatic view of single-layer electric slag welding process.**
(1) Electrode; (2) slag bath; (3) metal bath; (4) welded seam; (5) copper slides; (6) component.

P.T.F.E.-Impregnated Dry Bearings

By D. C. MITCHELL and A. E. BURKE*

FOR some considerable time it has been evident that a bearing material capable of operating without any liquid lubricant would fulfil a definite need. In addition, such a bearing should be capable of operation in the presence of unconventional liquids, and should be clean-running—an important consideration in such fields as food processing, textile and paper working, printing plants, and laundries.

A bearing which does not require a liquid lubricant can be expected to operate equally well at extremes of temperature and will be limited only by the mechanical properties of its constituent parts. A further point is that such a bearing can be left completely unserviced until the need for replacement arises, provided that it is chemically inert to the conditions in which it is operating.

These requirements are met by a new dry-bearing material based on the excellent anti-friction and wear properties of Polytetrafluoroethylene (P.T.F.E.),† incorporated into a porous metal matrix.

PROPERTIES OF P.T.F.E.

Polytetrafluoroethylene is a plastic polymer of the basic formula $(CF_2)_n$, and is a white, translucent solid at room temperatures; it can be obtained in three forms, i.e., as a solid, as a moulding powder, or as a dispersion in water. These three forms are chemically the same and differ in their physical form only. P.T.F.E. is extremely inert and is dissolved, attacked, or affected only by molten metals of the group sodium, potassium, etc., or by fluorine gas. No other materials have any detectable effect.

P.T.F.E. has a high softening point, of the order of 327° C; at this temperature it passes through a transition point which, but for the length of its long, straight, unbranched chains, would probably be its melting point. Like all plastics, it has poor thermal conductivity and a high coefficient of thermal expansion. Its bulk strength is rather low and for moulded materials a value of about 2000 psi is fairly normal. In addition, it has the tendency to cold-flow or deform plastically under very light loads[1].

As the result of a suggestion by Mr. Tait, of the Glacier Metal Co. Ltd., Shooter and Thomas[2], under Dr. Bowden of the Department of the Physical Chemistry of Rubbing Surfaces at Cambridge University, carried out a frictional investigation of this material and showed that it had a phenomenally low coefficient of friction, in the region of 0·04 to 0·05. These low values were obtained both between the plastic and any metal, and between two samples of the plastic.

In addition to confirming the good anti-friction properties of P.T.F.E., Bowden and his colleagues found that the plastic had excellent wear properties, as sliding friction appeared to take place at the surface of the plastic, as against the normal shearing process of the weaker material[3]. Any wear appeared to be due to the ploughing of the roughness of one member through the softer plastic.

CHOICE OF POROUS METAL FOR USE WITH P.T.F.E.

P.T.F.E. has certain disadvantages which limit its use in the solid form as a bearing material to very low-load and low-speed applications. These disadvantages are its poor thermal conductivity and high coefficient of thermal expansion, and its low strength, and susceptibility to cold flow. Furthermore, P.T.F.E. will deform over some considerable time under any residual internal stresses left in the plastic after a processing operation, so that machining of accurate shapes from this plastic is extremely difficult.

To overcome these disadvantages, it was proposed that P.T.F.E. be incorporated into a porous metal, so that the metal itself would provide the properties of good thermal conductivity, high strength, and dimensional stability, while the P.T.F.E. would provide the good anti-friction, chemical, and wear properties.

The choice of suitable metals is very large and limited only to those materials which can be handled by powder-metallurgy techniques to give a suitable porous structure. Some of those metals which have been investigated are copper, silver, iron, bronze, and stainless steel, but by far the most promising material appears to be that made by sintering spherical pre-alloyed bronze powder to a steel backing. This material is made by spreading thin layers of bronze powder, generally between 0·010 and 0·030 in. in thickness, onto copper-plated steel, the composite strip being passed through a sintering furnace at a controlled speed and temperature.

The resultant porous bronze is strongly adherent to the steel backing and has a porosity in the range of 20 to 50% and a bulk yield strength usually between 10,000 and 20,000 psi. The thin layers of bronze on the steel are particularly advantageous for bearing purposes, as the interconnected bronze structure will rapidly carry any heat generated at the bearing surface to the solid steel backing, which, furthermore, will give additional strength to the bronze lining. An additional point in favour of thin bronze linings is that they require less of the expensive P.T.F.E. to fill the voids within the bronze, thus reducing production costs.

METHODS OF IMPREGNATING METAL WITH P.T.F.E.

Of the several methods of making a composite structure of metal plus P.T.F.E.[4, 5, 6] the most important is that of impregnating a porous metal matrix with P.T.F.E. This impregnation can be carried out by two basic methods, depending on the form of the P.T.F.E. used. The first is an impregnation using high pressure plus high temperature with the solid polymer, while the second consists of using vacuum techniques with the dispersion of P.T.F.E. in water.

In the first method, which, contrary to the experience of Dr. Blainey[7], gave satisfactory results, a thin strip of the plastic is placed on the surface of the porous bronze in an enclosed die assembly and the whole is heated to a temperature between 330° and 400° C; a pressure of up to 30,000 psi is then applied to the surface of the plastic, thus forcing it into the porous material. The periods of pressure application vary between 30 seconds and 20 minutes, depending, amongst other factors, on the type of porous material being treated. The material which can be obtained by pressure impregnation is shown in Fig. 1, which is a photomicrograph of a heavily etched porous bronze impregnated with P.T.F.E. The dark metal and white P.T.F.E. standing proud of the metal are plainly visible.

In the second method of impregnation, the water dispersion of P.T.F.E. is used, containing in the most concentrated form 60% by weight or 40% by volume of P.T.F.E. The porous parts to be treated are placed in a sealed vessel containing the P.T.F.E. dispersion; by applying a vacuum of the order of 27 in. over the surface of the liquid and then releasing this vacuum, the liquid can be made to enter the pores of the metal. On removal from the dispersion, the part then contains both P.T.F.E. and the dispersing fluid, i.e., water. This is now dried in hot air to remove the water and to precipitate the P.T.F.E. within the metal. As the dispersion contains only 40% (i.e., two-fifths) by volume of P.T.F.E., only two-fifths of the available porosity will be filled; thus, for a material containing 30% voids initially, after treatment only 12% of the voids will

* Research Department, The Glacier Metal Co. Ltd.

† Known under the trade-names of " Fluon " (I.C.I.) and " Teflon " (du Pont de Nemours).

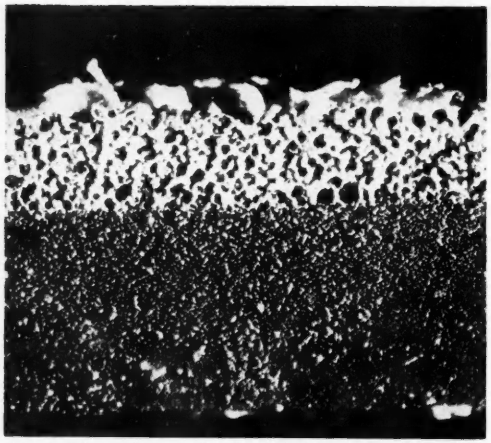

Fig. 1. **Photomicrograph of polished and heavily etched section of P.T.F.E.-impregnated bronze dry-bearing-material, showing distribution of plastic and porous metal** (× 20).

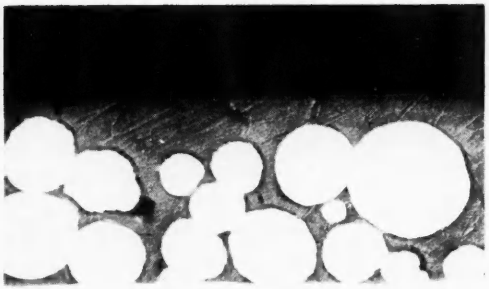

Fig. 2. **Photomicrograph of polished section of P.T.F.E. in bronze, showing the initial surface layer of P.T.F.E. on the surface of the bearing** (× 100).

be filled, leaving 18% still vacant. This therefore necessitates several successive treatments of filling and drying until a reasonable amount of the original 30% of voids has been filled; up to ten such treatments may be required. When all the available porosity has been filled, the part is cured at a temperature of about 360° C to fuse the particles of the plastic together and to produce a continuous plastic phase within the porous metal.

The final material produced by both methods is very similar, except that more P.T.F.E. can be introduced into a porous material by pressure impregnation than by vacuum impregnation. The structure produced consists of two completely interlocked sponge-like networks, one of the metal phase, say, bronze, and the other of the plastic P.T.F.E. In addition, both methods are applied in such a manner that a thin layer of pure P.T.F.E. is left over the whole surface of the porous metal; this P.T.F.E. generally has a thickness of between 0·0005 and 0·0015 in. A typical example of a pressure-impregnated material is shown in Fig. 2, in which the impregnated P.T.F.E. and surface layer of the plastic are clearly visible. It can be seen that the surface layer of pure P.T.F.E. is sufficiently thin to be amply supported by the underlying porous metal, and it will be shown that this surface layer serves a most important function during the initial bedding-in of the bearing.

The porous metals so treated can be in the shape of bearings before treatment, or parts can be made into bearings after impregnation. The pressure-impregnated material is slightly more limited with regard to the shapes that can be produced, because it can be applied only to porous parts in the flat form or to relatively simple and regular bearing forms, whereas parts of any shape or size can be treated by the vacuum technique.

This limitation to flat strip from pressure impregnation is not a serious disadvantage, as it can be processed into all the common forms of bearing, such as thrust washers, slide ways, bushes, half bearings, and (with special tooling) spherical bearings.

To demonstrate the type and range of materials now being produced, Fig. 3 shows a selection of bushes and thrust washers made from P.T.F.E.-impregnated bronze on steel strip.

Bearings are always manufactured to the finished size; the surface of the bearings cannot be machined in any way, nor is it possible to lap or scrape the bearing surface, as these operations would remove the initial surface layer of the bearing and cause premature failure when in service. Bushes and thrust washers are manufactured with sufficient accuracy to ensure that, provided the correct housing and shaft sizes are used, the bearing will be of the correct size to give good operating conditions.

Method of Operation of a P.T.F.E.-Impregnated Metal Bearing

Bearings of this type operate by maintaining a thin film of P.T.F.E. on one or both of the rubbing members in a bearing assembly. The frictional properties of a porous metal impregnated with P.T.F.E. are given in a paper by Bowden[8], who shows that the frictional properties obtained for the composite impregnated porous metal are the same as those of the solid plastic and that these good anti-friction properties are maintained after a considerable amount of rubbing, over a wide range of temperatures, has taken place between the test surfaces.

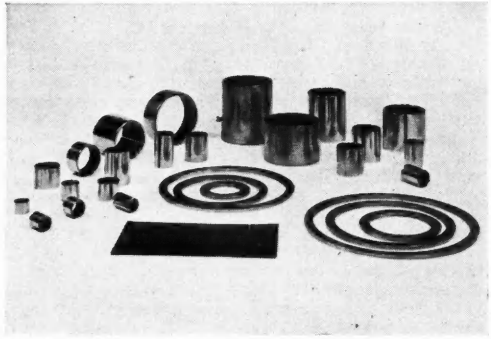

Fig. 3. **Selection of bushes and thrust washers made from P.T.F.E.-impregnated bronze on steel strip.**

As mentioned previously, the bearing has a layer of pure P.T.F.E. on the surface of two completely interlocked sponge networks, one of P.T.F.E. and the other of the supporting metal. In the initial stages of operation of the bearing assembly, the surface layer serves three purposes:—It ensures right from the start of the operation that the rubbing contact is between at least one surface of plastic and one of metal. Then, as the peaks of shaft roughness plough through the surface layer, the shaft itself becomes coated with a thin layer of P.T.F.E., which fills the hollows and rounds off the asperities on this metal mating member; there is,

however, a limit to the degree of roughness which can be tolerated in the mating metal surface. Finally, although this surface layer is comparatively thin, it is sufficient to accommodate some misalignment of the two members of the bearing assembly. Thus, the surface layer of pure plastic carries out a very valuable function and ensures that, after a few hours, the bearing is operating correctly. At the end of this running-in period, rubbing is taking place between two P.T.F.E. surfaces—one on the shaft and the other on the bearing.

The very thin film of P.T.F.E. remaining from the surface layer has to be maintained during the life of the bearing. This thin layer may be removed at some point on the bearing, permitting local metal-to-metal contact to take place, thereby producing a local hot spot, so that the heat generated at this point will cause the materials in that area to expand. The P.T.F.E., which has a much higher coefficient of expansion than the metal, will expand out of the pores of the porous matrix and will be wiped over the exposed metal surfaces, once again providing a continuous film of plastic over the bearing surface.

It can thus be appreciated that the limiting factor for a dry bearing of this type is the amount of work which is being done at the bearing surface, since this will determine the amount of damage being caused and the rate at which damaged areas have to be healed by expansion of the P.T.F.E. from the porous metal matrix. In addition, the more rapidly the heat generated at the hot spot is dissipated, the smaller will be the amount of P.T.F.E. exuded from the metal and consequently the longer will be the life of the bearing assembly. In other words, the life of a P.T.F.E.-impregnated porous-metal dry bearing is largely dependent on the amount of work being done in the bearing and the rate at which the heat energy liberated by this work can be dissipated. Hence, the load-carrying capacity of the material may here be expressed in terms of a factor PV, where P is the load pressure [psi] and V the surface speed [fpm] applied to the bearing material.

Testing Machines

For the purpose of testing thrust washers and bushes, special testing machines were designed. The essential features of the thrust-washer machine are the hydraulic loading ram which carries the test sleeve and which can be rotated at any of a series of preselected speeds, and a test head which is rigidly fixed to the base of the machine and which carries the test specimen, against which the sleeve in the loading ram abuts. The specimen is mounted so that it is free to align itself with the rotating sleeve.

For most testing, three speeds are used, i.e., 750, 1500, and 3000 rpm, equivalent to a mean surface speed of 243, 486, and 972 fpm on a thrust washer of 1·575 in. o.d. and 0·905 in. i.d. The load applied to the test assembly is controlled by means of a relief valve in the hydraulic circuit which supplies the pressure to the loading ram. The pressure in the hydraulic loading cylinder is recorded on a standard bourdon-type pressure gauge which is frequently calibrated. Any load of up to a maximum of 250 lb. can be applied to the test assembly. This machine is designed for the continuous recording of the thrust-washer temperature by means of a thermocouple in contact with the back of the test specimen. The overall wear of the test assembly at any instant can be obtained with a dial gauge which measures the rise of the loading ram as wear takes place between the thrust washer and sleeve.

The essential parts of the machine for testing bushes are a test shaft which is mounted in the shaft housing and which can be rotated at any speed, and the bush housing, which is held in the test head, to which loads in the form of dead weights are applied.

This machine tests bushes of from ¼-in. bore up to 1-in. bore, with a unidirectional load. For most tests, speeds of 250, 750, 1500, and 3000 rpm are used, with loads up to a maximum of 150 lb. The machine is designed for the continuous recording of bearing temperature by means of a thermocouple in contact with the back of the test bush in the middle of the loaded area. The overall wear of the assembly is obtained at any instant by measuring the drop of the test housing as wear takes place in the bearing assembly.

Test Procedure

The test bush or thrust washer is manufactured from the appropriate material, using standard engineering technique and taking care not to damage or remove the surface layer of P.T.F.E. The mating surfaces for the test materials are manufactured from mild steel, unless otherwise stated, using normal engineering techniques up to the final preparation stage of the test surface. In the case of thrust-washer sleeves, the final surface is obtained by surface grinding and then lapping with fine-grade emery paste. For bush shafts the final surface is obtained by fine-turning and then polishing with 0-grade emery, to remove the turning marks.

All parts to be tested, sleeves, shafts, bushes, and thrust washers, are degreased prior to use by boiling in trichloroethylene, and are then mounted in the appropriate test machine with great care, to ensure that no oil or other unwanted material reaches the test surfaces.

Before testing, all parts are carefully measured and weighed, so that the final wear figures can be accurately ascertained. The parts are set into the appropriate test machine, the correct test speed is selected, and the test load determined.

Tests are started at the full value of test conditions for each run and an initial reading is immediately taken with wear-measuring instruments. This reading is the datum for all further wear readings, to determine the overall wear in the assembly from the start of a test. Wear readings are taken every 15 to 30 min for the first few hours of testing and, provided that the readings are satisfactory, wear values are then taken only three times a day.

A bearing assembly is considered to have failed if any one of three conditions is present:—

(1) A total wear in excess of 0·005 in.
(2) A temperature in excess of 300° C.
(3) Seizure of the test assembly.

Tests are generally not continued beyond 1000 hours. This latter criterion is used because there is evidence that tests which have run for this period will continue almost indefinitely and test capacity is too valuable to be tied to a few tests running for long periods. However, several tests have been allowed to run for more than 1000 hours, and Table I is an example of two such tests. In each case, the specimens were still running at the end of the stated time but, as test heads were required, the tests were terminated.

Table I.

Type of test	Load, psi	Speed, rpm	PV-value	Period, hr	Total wear at end of test, in.
Bush	15	1500	3700	1637	0·0022
Thrust washer	40	750	9720	2912	0·0032

The stopping of tests after 1000 hours has the disadvantage that in test programmes for which some tests run 1000 hours and other tests do not, the average test life is always on the low side compared with what it should be if the 1000-hr tests were allowed to continue.

An interesting point concerning tests which run for protracted periods and which are examined prior to failure, is the greenish tinge which both bearing members take on; this discolouration, which is a good guide to a correctly running bearing, is thought to be due to fluorides of the metals present, formed when fluorine is produced during the thermal degradation of the P.T.F.E. at local hot spots.

By far the most common cause of failure is a wear

in excess of 0·005 in., though, in some tests at low speed and low load, 1000-hr tests may predominate.

General Test Results

Tests have been carried out under a wide variety of test conditions on many materials made with various modifications to the main methods of impregnation. The results which are discussed are those which have been influenced by good and bad impregnation conditions, and in which the tests have been carried out without any aids to increase test life, as, for instance, by cooling the test assembly or bedding-in the specimens. The results can therefore be considered to be rather pessimistic for the range of standard materials (Glacier Dry Bearings, DP series) being produced, as production conditions for this range will be at their best possible levels within the limits of present knowledge.

It has been stated previously that the life of a dry bearing assembly is determined by the amount of work that is being done at the bearing surface. This means that the life of the bearing will be equally dependent on the pressure P and surface speed V of the rubbing surfaces. It is the joint effect of P and V which determines the bearing life and not either of these factors separately. This means that, for any value of P and of V such that PV is constant, the bearing assembly will have a constant life expectancy.

The error obtained in bearing tests is always rather high because of uncontrollable conditions affecting bearing performance, and it is found that the error for a particular test condition is proportional to the length of the test; consequently, to make the error reasonably uniform for all tests, it is convenient to consider the logarithm of the hours to failure. The results of tests on dry-bearing materials are therefore considered in the form of the $\log_{10}$ hours to failure and the $\log_{10} PV$ for a given test. This then gives an approximately linear relationship between the PV-factor and the hours to failure of a bearing.

Tests on Thrust Washers

The results of some work on thrust washers are given in Fig. 4, which is a logarithmic plot of the test life in hours against the PV-value used for each series of tests. Each point plotted, which is the mean of the logarithms of the test life in hours, represents a test programme or part of a test programme, in which various impregnation conditions may have been used. The number of results which have been measured to give the point plotted is indicated beside the point.

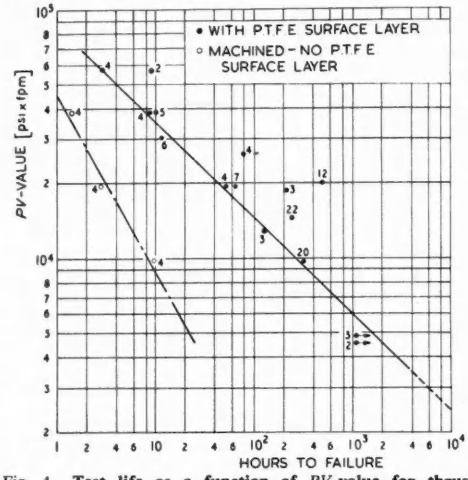

Fig. 4. **Test life as a function of PV-value for thrust washers.**

The line drawn in connection with these results is not the best line through the points given, but is a line which it is felt can be safely used for design purposes based on the results. It can be seen that, for a life of 1000 hr, a PV-value of about 6000 can be used, and by extrapolation of the results it seems very probable that, if a life of about 10,000 hr is required, a PV-value of about 2500 should apply.

The effect of machining of the dry-bearing materials on test life can be gauged from the other series of points plotted in Fig. 4. This shows the results of a series of tests in which the surface layer of P.T.F.E. present on the normal dry-bearing material was machined away prior to testing. The disastrous effects caused by this machining are only too evident from the test results. As the materials were originally identical, this difference can therefore be attributed to the surface layer of P.T.F.E. on the thrust washers and to the great advantages which this layer gives during the initial running-in stages of the bearing assembly.

Effect of Mating Material on Thrust-Washer Performance

It has been found that P.T.F.E. in bronze will operate very successfully against a wide range of materials. However, the majority of the work has been effected with a plain mild steel with a hardness of 140 to 160 V.P.N.; in fact, all the results given are for tests run against this mild steel.

To show that mild steel is as effective as a hardened steel for use with P.T.F.E. bearings, Table II has been compiled. The soft mild-steel shafts had a hardness of about 150 V.P.N., while the hard-steel shafts had a hardness of about 710 V.P.N.

Table II.

No. of tests	PV-value	Type of shaft	Mean $\log_{10}$ hours to failure
1	9970	Hard steel	2·62
1	9970	Soft steel	2·06
2	19,940	Hard steel	2·40
3	19,940	Soft steel	2·36
1	39,880	Hard steel	0·40
1	39,880	Soft steel	0·70

It can be established from these results that there is no significant difference between tests carried out with mild steel and those with hardened steel. This result is most advantageous, as it means that, when these dry bearings are applied to industrial applications, the shaft material for use with the bearing can be of any specification suitable to the user.

Comparative Thrust-Washer Tests on Materials Similar in Application to P.T.F.E. in Bronze

Many different materials have been investigated to find a suitable bearing material which will operate under the conditions specified previously; in addition, materials which partly meet the specified conditions have been tested. Some of the more interesting results are given in Table III, in which the PV-value used for testing was about 18,000.

Table III.

Type of material	No. of tests	Hours to failure	Cause of failure
P.T.F.E. in bronze on steel	3	213	0·005-in. wear
Graphite (solid) ..	2	54	0·010-in. wear
Oil-impregnated porous bronze ..	3	29	Disintegration
Nylon in bronze on steel	2	7	Excess temperature and wear
Solid P.T.F.E. ..	1	0·8	Disintegration
Solid nylon	1	0·1	Disintegration

This table indicates that, for the materials tested, the P.T.F.E.-impregnated bronze on steel is superior to the other materials under thrust-washer conditions.

Tests on Bushes

The results of some work on bushes are given in

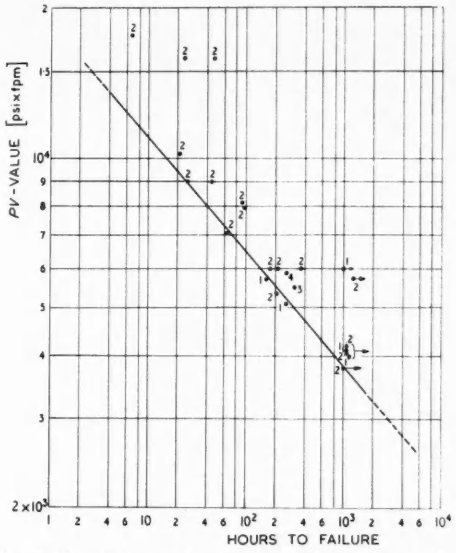

Fig. 5. **Test life as a function of *PV*-value for bushes.**

Fig. 5, which is a logarithmic plot of the hours to failure against *PV*-value. The *PV*-values are obtained with several combinations of load and speed, and each point plotted is the mean value of several tests on materials which may have been made by various modifications to the impregnation process. The actual number of tests which have been averaged is indicated beside each point.

The line drawn in connection with these points is once again not the best line through the values but one which it is felt can safely be used for design purposes with these dry bearings. It can be seen that a life in excess of 1000 hr will be obtained with a *PV*-value of about 3700, and by extrapolation of the results it seems probable that a life in excess of 10,000 hr can be anticipated with a *PV*-value of 2300.

The lower *PV*-value to give 1000-hr life for bushes, compared with thrust washers, is due to the different geometric conditions under which the bearings are operating. For thrust washers, the whole area of the bearing surface is used, whereas, for bushes tested under these conditions, the test load and consequent wear are concentrated in only one small area of the bush. Theoretically, under these conditions, there is only line contact between bush and shaft but, owing to elastic deflections and, more important, to bedding-in of bush and shaft, the loaded area is spread around about 90 deg. of arc of the bush; on the other hand, for design purposes, the full projected wear of the bush is used to determine the load applied to the bearing. Even within this area of wear, the load is probably not evenly distributed, the majority being carried under the point of load application.

If the slopes of the lines relating bush and thrust-washer results are considered, it will be found that these are in reasonable agreement, confirming that the mechanism of operation is similar for both cases.

It should be noted that somewhat lower *PV*-values than those quoted above are recommended for general application, to allow for possible unfavourable circumstances, e.g., poor heat dissipation from the bearing assembly.

Wear of Bearings during Test

The wear of bearings whilst under test follows a fairly consistent pattern. In the first few hours of operation, the surface layer of pure P.T.F.E. is worn away and, as a result, a very thin coating of plastic is left on both members in the assembly. The initial surface layer is about 0·0005 to 0·0015 in. and, as this is used up, a wear of up to 0·0015 in. is recorded. When this running-in process is complete, the bearing settles down to the normal method of operation and no further detectable wear takes place until the bearing begins to fail. The onset of failure is shown by an increase in wear to such an extent that, in a few hours, a wear of up to 0·005 in. is recorded, when it is assumed that the bearing has failed and the test is stopped.

Typical examples of wear curves for bushes and thrust washers are given in Fig. 6, in which the total wear of the bearing assembly is plotted against time. The initial bedding-in of the thrust washer is about 0·0007 in. and for the bush 0·0015 in., after which both continue to run with no increase in wear until just prior to failure. This wear pattern is followed by all the dry bearings under test, except that tests which run for 1000 hours obviously do not show the rapid increase in wear associated with failure. There is some slight difference between the wear pattern of thrust washers and bushes in that, owing to the concentration of load with the bushes tested under the conditions used, these generally exhibit a slightly higher initial wear than thrust washers. Comparison of the two results shown in Fig. 6 illustrates this point.

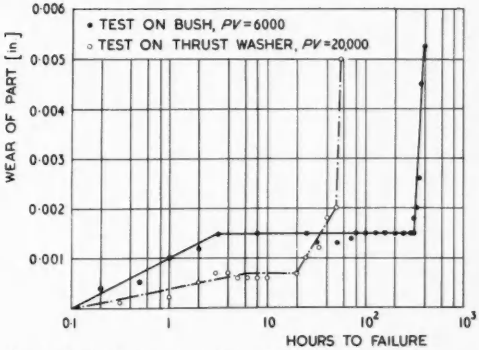

Fig. 6. **Typical wear curves for P.T.F.E.-impregnated bronze.**

When a test has failed as a result of wear, and the relative wear of the two components is studied, it is generally found that about 0·004-in. wear has taken place on the bearing, and about 0·001-in. wear on the mating member.

Temperature of Bearing when Tested at Various *PV*-Values

The temperatures obtained during testing are very much dependent on the physical set-up of the test machine. The main value of the temperatures recorded during testing is to show that the bearing is operating correctly, and to give evidence of the mechanism of operation of dry bearings.

The relationship between test temperature and *PV*-value should be linear, if the coefficient of friction for the bearing assembly is independent of load and speed. Fig. 7 shows curve *B*, relating *PV*-values with the mean bearing temperature for a series of tests on bushes. Each point plotted on the graph is the mean of several tests in a test programme, and the *PV*-values are obtained from various combinations of load and speed. This graph shows that, within experimental error, the relationship between stable test temperature and *PV* is linear, and therefore the mechanism is that postulated above and in keeping with the laws of dry friction propounded by Bowden & Tabor, *et al*[9]. The results also confirm that the amount of work being done in the

bearing assembly is equally affected by both bearing pressure and the surface speed of the moving member.

For thrust washers, a similar result is obtained as for tests on bushes, but the temperatures recorded with thrust washers are somewhat higher for the same *PV*-values. This is because the test head is more enclosed, and there is some transfer of heat from the hydraulic oil in the loading cylinder of the test rig used.

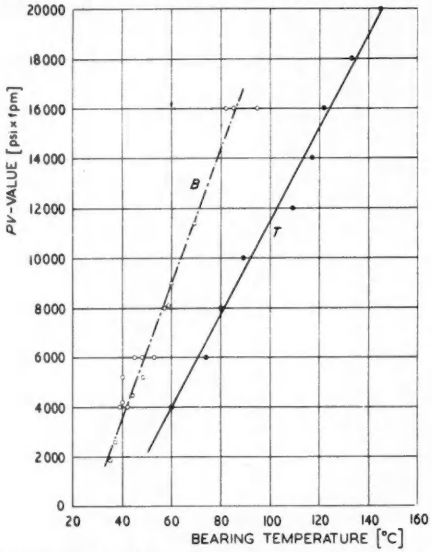

Fig. 7. **Mean bearing temperature as a function of *PV*-value for a series of tests on bushes (curve *B*), and one test on a thrust washer (curve *T*).**
Ambient temperature = 26° C.

Curve *T* in Fig. 7 shows the results of tests on one thrust washer at a series of *PV*-values, and demonstrates that individual bearings have the same relationship between *PV* and temperature as that obtained with several different bearings tested at various *PV*-values.

Effect of Surface Finish on Bearing Performance

It might appear that the surface finish applied to the mating member for a dry bearing is critical because of the ploughing effect to be expected with a poor surface finish. This is partly true, but is not nearly as serious as might be expected, because of the surface layer of P.T.F.E., which rapidly smooths out the roughness on the mating surface, giving a good bearing combination.

Table IV shows the results of a series of tests on bushes with a fine-turned shaft, and on similar bushes with similar shafts, except that, prior to testing, the turning marks were removed with 0-grade emery paper. Some difference between the two surface finishes are noted, but the difference is not as great as would be expected between the two types of surface used.

Table IV.

No. of tests	Shaft finish	*PV*-value	Mean $\log_{10}$ hours to end of test	Remarks
1	Fine-turned	4000	2·17	Failure by wear of 0·005 in.
1	Fine-turned and polished	4000	3·01+	Not failed; stopped after 1000 hr
2	Fine-turned	6000	2·38	Failure by wear of 0·005 in.
1	Fine-turned and polished	6000	3·00+	Not failed; stopped after 1000 hr

These results are confirmed by a series of tests carried out with thrust washers, using various surface finishes. Results are given in Table V. All the mild-steel sleeves used for the test were first finished, using the same fine-grinding technique. The sleeves to be used for the lapped surface were then selected and finished on a surface plate with a fine-grade lapping paste; the sleeves to be polished were first taken through the normal range of emery polishing paper down to 000-grade, and were then given a final polish with fine aluminium on a Selvyt cloth.

Table V.

No. of tests	Surface finish	*PV*-value	Mean $\log_{10}$ hours to failure
4	Fine-ground	20,400	2·65
5	Ground and lapped	20,400	2·72
5	Ground and polished	20,400	2·52

These results indicate that, providing the surface finish is of a reasonable nature, the surface layer of pure P.T.F.E. will be sufficient to give the same test performance, regardless of the surface finish.

Effect of Liquids on Performance of P.T.F.E. in Bronze

Some little work has been carried out with liquids flowing around or through the P.T.F.E. in bronze bearings. The presence of a liquid of any type in or around the bearing can be extremely beneficial. This liquid will act as a means of removing the heat from the bearing surface in the same way as lubricant in normal oil-lubricated bearings. Thus, much higher loads at much higher speeds can be carried by the bearing. In addition to acting as a means of removing the heat generated in the bearing, the presence of a liquid will also lead to hydrodynamic lubrication, with a consequent further increase in load-carrying capacity of the assembly. An excellent illustration of this increased load-carrying capacity is given in a paper presented to the Iron & Steel Symposium on Powder Metallurgy by Dr. Blainey[7]. Tests were carried out on P.T.F.E.-impregnated bronze journal bearings running with a full supply of gasoline, and runs of many hundreds of hours were obtained with *PV*-values of up to 40,000. It is unfortunate, however, that these tests were carried out with materials which were rather low in P.T.F.E. content and with materials from which the surface layer of P.T.F.E. had been removed. These factors may explain the poor results obtained by Dr. Blainey under dry conditions.

The authors' experience with fluids flowing around the bearing has been limited to trace-quantities of lubricants, which have been shown to improve most markedly the life of the materials; it should, however, be understood that these bearings have been primarily designed for satisfactory operation under completely dry conditions.

Acknowledgements:—

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Mechanical Tests on Rubber Sealing Rings for Rotating Shafts

By J. LEIN. (From *Konstruktion*, Vol. 6, No. 10, October 1954, pp. 384-389, 10 illustrations.)

SEALING rings made of synthetic rubber, for sealing the interior of a housing against leakage along a rotating shaft, have proved successful even at high rotational speeds, and with differences in inside and outside pressures. The synthetic rubber ring is pressed into a steel frame, force-fitted into a flange of the housing to be sealed. The radial pressure exerted by the ring on the rotating shaft is due partly to the hoop stresses in the rubber, and partly to the elastic force of the expanded circumferential spring (Fig. 1).

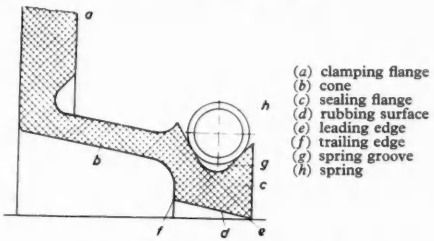

Fig. 1. **Typical design of a sealing ring.**

In the testing arrangement used, an oscillating housing is freely supported on ball bearings, prevented from rotating by a weighing arm, and filled with oil up to the centre line. The oil can be heated or cooled. The ring seals the oil-filled housing against a rotating trunnion of 80 mm in diameter, hardened and ground, driven by a variable-speed motor. A thermometer in the housing indicates the oil temperature near the surface of the trunnion, while another thermometer, rotating with the trunnion, gives the trunnion temperature and can be read during tests. The radial pressure acting on the trunnion is measured before and after each test on a special attachment to the test bench. The amount of oil leaking out of the housing is also measured during tests.

If R is the total distributed radial force acting over the whole circumference of the trunnion, U the total tangential frictional force, T the friction torque indicated by the weighing arm, r the trunnion radius, and μ the coefficient of friction, then

$$T = Ur = \mu Rr.$$

RESULTS OF TESTS

(1) *Qualitative results and service experience:*—

(a) Rings of synthetic rubber are superior to leather rings as regards maximum service temperature, sealing efficiency, durability, and friction. Leather is still being used for rubbing speeds up to 6 m/sec and temperatures up to 80° C, especially for temperatures below freezing point. Synthetic rubber withstands temperatures up to 120° C and is suitable for any rubbing speed which does not increase the temperature beyond this limit, i.e., about 12 to 15 m/sec for conventional designs, provided that some oil always reaches the ring and that the rubbing surfaces are not heated by other sources. Higher speeds require artificial cooling by circulating oil or oil spray, and an economic limit is reached at about 20 to 25 m/sec; however, this limit also applies to sliding and rolling bearings and to the use of lubricating oils. Rings made of silicone-base materials may be expected to withstand higher temperatures, up to about 180° C.

(b) The scatter of test results always obliges the testing of a number of rings under identical conditions. The rubber is never quite homogeneous and has wide manufacturing tolerances, the fitting differs from ring to ring, and running tests introduce continuous changes due to temperature, oxidation, and lubrication effects, so that only average values can be given for life, sealing efficiency, and friction torque. The probability of a particular ring reaching these average values is not higher than 90%, so that even preliminary tests should not be undertaken on less than 5 to 10 rings under identical conditions.

(c) Within certain limits (0·05 mm eccentricity of trunnion at 3000 rpm), eccentricity of shaft rotation has no influence on sealing efficiency and friction torque. Thus, the limits imposed by other considerations on frequency and amplitude of eccentric shaft motion can usually be accommodated by this type of sealing ring. Friction and sealing power also remain unaffected by any static eccentricity of the shaft axis in relation to the axis of the ring, provided that this eccentricity is less than 1% of the shaft diameter, or 2% for small shafts. Since the diameter of the leading edge of the ring must be smaller than the shaft diameter, so that sufficient radial pressure can be guaranteed in spite of unfavourable manufacturing tolerances, the radial pressure on shafts of nominal diameter will be larger than required and the ring will work satisfactorily even with shafts below nominal size. During running, the ring usually grinds a groove of up to 0·1 mm depth in the shaft and, if a new ring is fitted, the shaft diameter may be safely ground down correspondingly, without any ill effects on subsequent running.

(d) If the roughness of the trunnion surface is about 2 to 3 μ, surface roughness has no further influence on sealing efficiency and friction torque. The life of the trunnion, however, depends on the wear resistance of its surface. Surface-finish tolerances may be kept fairly wide, because the ring very rapidly creates a mirror finish, even on a hardened shaft, and polishes roughnesses to below 1 μ. Ordinary grinding is therefore quite sufficient. However, although the ring itself does not wear, its abrasive effect on the shaft is such that the shaft surface must be hardened ($R_C = 60$), or even fine-ground and hard-chromium plated for speeds above 15 m/sec. Metals of lower wear resistance, such as unhardened steel, light alloys, and cast iron, can be used only for very low speeds. Axial feed during turning or grinding must be as small as possible; otherwise, the helix may draw oil along the shaft. However, if the shaft rotates in one direction only, this effect may be utilised to force oil back into the housing.

(e) Sealing efficiency and friction torque depend on the type of oil used. The variety and rapid development of oils in common use make it impossible to arrive at any general conclusions, but it appears that some oils lead to higher demands on the ring and to the development of new types of rubber and to improved shapes of sealing flange.

(f) With certain types of rubber, the percentage of leak-proof rings is higher when the friction torque is small. The serious effect of machining helices proves that the oil film between ring and shaft is not cut off by the sealing edge but persists in spite of high radial pressure. Hence, an excessive pressure serves only to increase friction torque, shaft wear, and temperature, and causes premature leakage and destruction of the ring. The coefficient of friction μ is usually between 0·2 and 0·3, and the manufacturer of the sealing ring chooses the radial pressure so as to obtain an optimum sealing effect and friction torque. The circumferential spring must never be removed.

Chemico-physical phenomena, such as adsorption, surface tension, wetting, and capillary action, as well as mechanical phenomena such as oscillations, damping, and change in radial pressure, may contribute to friction

and leakage. When a new ring is fitted, rubber and spring are expanded elastically, so that a radial pressure is generated. As, however, the elastic resistance of the ring changes appreciably during running, the radial pressure will be subjected to unpredictable fluctuations. Nevertheless, tests with stationary trunnions, have shown that the pressure remains uniform over the circumference and is high enough to prevent egress of oil, even at elevated temperatures. An oil film is formed on the trunnion outside the housing, but no further oil leaks along the contact faces. It may be assumed that, during running, sufficient radial pressure is maintained to close any temporary gap; hence, leakage of oil can be due only to such momentary gaps caused by relative motion or vibrations. Friction may cause vibrations of the sealing flange in the circumferential direction, and eccentric running of the shaft may cause radial vibrations. However, calculation has shown that 1% of the actual radial pressure would prevent any loss of contact in radial vibrations, an indirect proof that a higher value of radial pressure is necessary in practice, not because of possible eccentricity but because of capillary and other boundary forces.

(2) *Properties of the rubber:*—

The typical course of deformation of a testpiece of soft rubber, 20 mm in height and 20 mm in diameter, under a maintained uniaxial compressive load of 10 kg/cm² and after unloading, is illustrated in Fig. 2. Fig. 3 shows the variation with temperature of the modulus of elasticity E, calculated from the reduction of height reached 1 sec after loading, of the elastic after-effect or "flow" measured 1 hour after loading, and of the permanent set measured 1 hour after unloading, expressed as a percentage of the total deflection. The dynamic load/deflection diagram under a fluctuating load is indicated schematically in Fig. 4 and leads to the definition of the dynamic stiffness as $C = \Delta P/\Delta H$ and of the damping coefficient as D = energy lost/energy applied $= (F_2/F_1) \times 100\%$; the corresponding change with temperature is given in Fig. 5. The deflection of rubber under a given static load thus reaches its final value only after a protracted period, or inversely, if the rubber is given a constant deflection, the stress decreases asymptotically towards a minimum value. Hence, the shorter the time or the higher the frequency of loading, the higher is the modulus E of rubber. If exposed to temperatures at or above 100° C, the rubber is increasingly vulcanised and modulus E rises. On the other hand, rubber swells and softens in oil, especially at higher temperatures, so that the increase in E is partly compensated. As swelling is favoured by tensile stresses and impeded by compressive stresses in the rubber, the spring serves not only to increase the radial pressure of the sealing ring but also to prevent by compression any excessive absorption of oil. Absorption of oil does not influence the damping capacity of the rubber, but storage of rubber in hot air gradually reduces its strength.

(3) *Radial pressure:*—

With the great majority of rings tested, the initial radial pressure of about 6·28 kg total force over the whole circumference did not change appreciably after 100 hours continuous running in Mobiloil "Arctic" oil. Any actual changes depended mainly on temperature and time. Rings subjected to high friction harden not only by gradual vulcanisation but also because the heat generated first softens the rubber, so that the contact area rapidly spreads over the whole width of the sealing flange; the entire spring pressure then acts on this flange and increases the radial pressure, giving rise to the friction and vulcanising effect. However, if the entire spring load is transmitted to the trunnion, radial pressure cannot increase in this way, and then changes only because of fatigue of the spring, which may lose 10% of its initial tension. Loss in pressure caused by the spring cutting into the rubber may be ignored.

(4) *Friction and torque:*—

The average friction torque for a sealing ring of 80 mm diameter and 6 kg total radial pressure is 6 to 7 kgcm. As the total radial force by definition depends only on the expansion of sealing flange and spring, and as in smaller rings the effect of the relatively larger expansion is compensated by the smaller cross-section, the radial force is about equal for rings of all sizes, and friction torque is therefore roughly proportional to the shaft diameter. If friction is high, the torque may change appreciably during running, even at constant speed and temperature; small friction torques change only slowly and by small amounts.

Several independent tests have shown that the total friction consists of boundary friction, which is independent of speed, and of hydrodynamic friction, which varies with speed; at 3000 rpm both these factors are about equal. With boundary friction, the trunnion and ring are covered by a very thin adherent oil film, which does not permit any general hydrodynamic shear of oil molecules. Since the surface roughness of the trunnion is about 3 μ, and that of the sealing ring 3 to 5 μ, it is likely that in service there exists rather a mixture of hydrodynamic, boundary, and dry friction, the latter occurring at the roughness peaks. Eccentric running of the trunnion possibly exerts a pumping effect and creates a weak film, but more probably the capillary forces draw some oil into the narrow gap between sealing flange and trunnion and create a hydrodynamic film. However, since this film cannot sustain any load because of the absence of a circumferential oil wedge, the radial pressure must be taken up by the roughness peaks, with a mixture of boundary and dry friction. That hydrodynamic friction is present has been proved

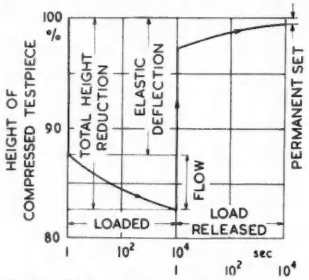

Fig. 2. **Effect of time on the compression of a soft rubber testpiece.**

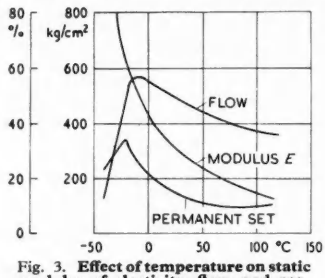

Fig. 3. **Effect of temperature on static modulus of elasticity, flow, and permanent set of soft rubber.**

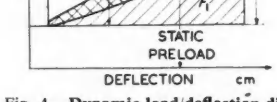

Fig. 4. **Dynamic load/deflection diagram of soft rubber.**

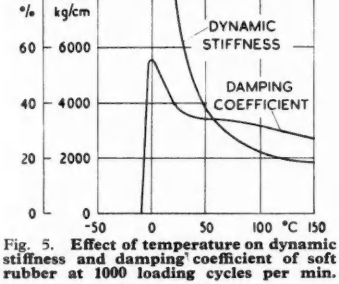

Fig. 5. **Effect of temperature on dynamic stiffness and damping coefficient of soft rubber at 1000 loading cycles per min.**

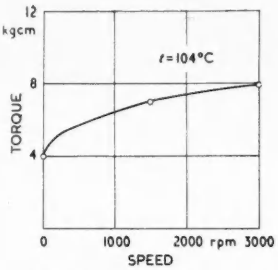

Fig. 6. **Total friction torque as a function of speed.**

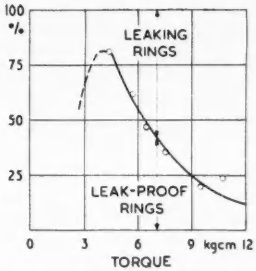

Fig. 7. **Percentage of leak-proof and leaking rubber rings as a function of average friction torque.**

by continuous recording of friction torque. When speed is reduced by a certain amount and then kept constant, torque first decreases but then increases again, because the oil film is cooler at the lower speed. The opposite holds true when speed is increased.

Total torque varies with speed, as indicated in Fig. 6. Tests on 700 rings made of a graphitised Perbuna mixture have shown that the percentage of leak-proof rings reaches a pronounced maximum when the average friction torque is 4 to 5 kgcm. Rings with an average friction torque of 12 kgcm are seldom leak-proof (Fig. 7).

(5) *Oscillations:—*

Since the trunnion always runs eccentrically, even if only by a very small amount, a relative motion exists between trunnion and sealing flange. This motion was observed microscopically and by cathode-ray traces taken from a capacitance pick-up consisting of the trunnion surface and a small copper plate embedded in the ring. A standard type of sealing flange was found to suffer a flexural vibration about the leading edge and a longitudinal vibration in the circumferential direction. The greater the radial pressure, the smaller is the amplitude of the bending vibration. The leading edge does not lift from the trunnion surface. In general, the amplitude of the longitudinal vibration is smaller than the eccentricity of the trunnion. Both vibrations have the same frequency as the rotation and do not influence the sealing power of the ring. Longitudinal vibrations are the main cause of the axial cracks frequently seen in the rubbing surfaces of rings which have been severely overstressed by friction and high temperature.

GERMANY

Current Trends in the Design of Steam Generators for Very High Pressures and Temperatures

By R. MUTKE. (From *Elektrizitätswirtschaft*, Vol. 53, No. 21, November 5, 1954, pp. 680-683, 6 illustrations.)

IN the following discussion, the author proposes to designate as " very high pressure " a range of from about 1850 psi to 2150 psi, now currently specified for steam generators with natural circulation, and to designate as " super-pressure " those pressures specified for steam generators working above the critical point. Similarly, with steam temperatures reaching the 1200° F mark, as specified for at least one German plant now under construction, it is proposed to summarise under the term " very high temperature " the range from about 990° F upwards, this range being possible only with the use of austenitic materials for all heat-exposed boiler parts.

The following table is indicative of modern German trends in boiler-design. This tabulated percentage analysis shows the heat requirements of each phase and indicates clearly that, with increasing pressures and temperatures, the superheat fraction gradually predominates, whilst the latent heat fraction decreases until, at the critical point, it will be found that evaporation proper vanishes.

To satisfy these conditions, the designer is confronted with the problem of suitably arranging superheater surfaces, which nowadays, by proportion, tend to increase considerably and which unavoidably obtrude into the region of higher flue-gas temperatures; as a result, they largely utilise radiant heat and improve superheater characteristics, at the expense, however, of being subjected to much more exacting working conditions. Furthermore, starting up of the plant will require even more attention than with convection-type superheaters located further back in the gas stream.

TABLE I. DISTRIBUTION OF HEAT INPUT WITH INCREASING PRESSURES AND TEMPERATURES.

Group	Type	Circulation	Steam Conditions		Feed-water	Reheat Conditions	Approximate breakdown of total heat, %			
			psi	°F	°F		Sensible heat	Latent heat	Super-heat	Reheat
A	Small plants only	Natural	450	750	212	—	20	65	15	—
B	Base load, large unit power stations, with re-circulated or fresh-water cooling	Natural	1930	990	390	—	25	45	30	—
C	As for " B "	Natural	1930	990	465	90% of steam at 430 psi and from 690 to 960° F	16	44	28	12
D	Super-pressure plant now under construction	Forced	4270	1120	635	1st reheat :—90% of steam at 1780 psi and from 850 to 1040° F 2nd reheat :—75% of steam at 430 psi and from 751 to 985° F	16	—	61	9 + 14

It has been found advantageous to utilise the less intensive non-luminous radiation from the furnace gases, rather than to expose superheater tubes to direct flame radiation; an example of an appropriate construction is illustrated in Fig. 1 which shows a boiler designed for the conditions of Group " B " in Table I and which uses a two-stage furnace, the first stage being cooled by water tubes and the second stage lined with radiant superheater surface.

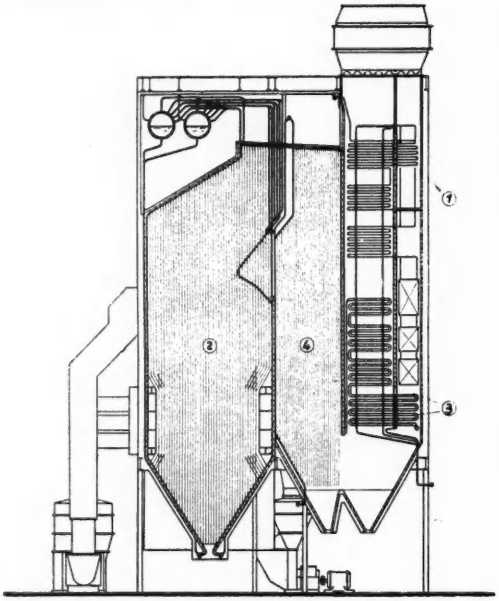

Fig. 1. **Radiant-heat steam generator for lignite.**
Evaporation 800,000 lb/hr ; 1940 psi ; 985° F.
(1) Economiser.
(2) Tube walls.
(3) Convection superheater.
(4) Radiant superheater.

Fig. 2 shows a steam generator with double wet-bottom furnace designed for the conditions of Group " C " in the table. It is fitted with superheaters directly built into the radiation space of the boiler, yet not exposed to direct flame radiation, combustion being completed in the furnace bottom.

The reheater is located between the primary and secondary sections of the convection superheater. Steam temperature is controlled by means of flue-gas dampers at the entry to the air heater. When load falls off, progressive closure of the dampers of the narrow flue-gas by-pass duct, baffled off from the last straight-through flue-gas pass, will divert a larger flue-gas volume into the convection superheaters. A similar type of temperature control is used for the boiler shown in Fig. 1.

Fig. 3 illustrates a steam generator designed for super-pressures according to the conditions of Group " D " in the table. At these pressures forced circulation must be introduced, and the liquid phase will transform steadily and with a continuous temperature rise into the superheat phase. The installed heat-transfer surface is mainly composed of superheaters.

The feedwater enters the economiser at about 635° F, leaving at 660° F; it then passes through the tubes lining the furnace, whence it is led to a demineralisation zone located in the last flue-gas pass. It is assumed that, even under conditions of super-pressure, the steam passes through a phase of demineralisation before entering the superheat phase, where the temperature is raised from about 770° F to 790° F. From here onwards all pressure parts of the boiler consist of superheaters only, starting from the neck of the combustion-chamber. Radiant superheater surfaces form the walls of the first and second stage of the furnace, whilst a convection superheater is inserted into the narrow flue-gas by-pass duct of the type previously described. It will be realised that with forced circulation the boiler outlet temperature remains independent of load variations; this does not apply to the reheat temperatures, which follow the pattern of the load variations, on account of the pure convection-type surfaces located in the last straight-through flue-gas pass. Progressive closure of the outlet dampers of the by-pass duct will direct a larger flue-gas volume over the interlaced reheaters, thus keeping the temperature constant, with the additional use of spray-cooling, if necessary. This damper-control action simultaneously reduces heat transfer through the convection superheater in the by-pass duct. Thus it counteracts the effect of the relative increase in radiant superheater surface, which, at reduced loads, tends to drift back from its initial zone towards the water end of the furnace, owing to the increased absorption of radiant heat in the furnace.

Boiler-steam outlet temperature may be controlled by the use of two mixing headers housed under the boiler roof for spray-cooling.

Superheater Design

It is evident that the superheater is fast becoming the predominant component of the boiler pressure-system. The greatest care must be given to its design, especially with regard to working temperatures and the correlated mechanical properties of the tube material. German standards fix the temperature difference between the outer skin of the tube and the mean steam temperature at maximum continuous rating at 72° F for convection heaters, and at 117° F for radiant heaters, though narrower margins are acceptable, if justified by thermo-

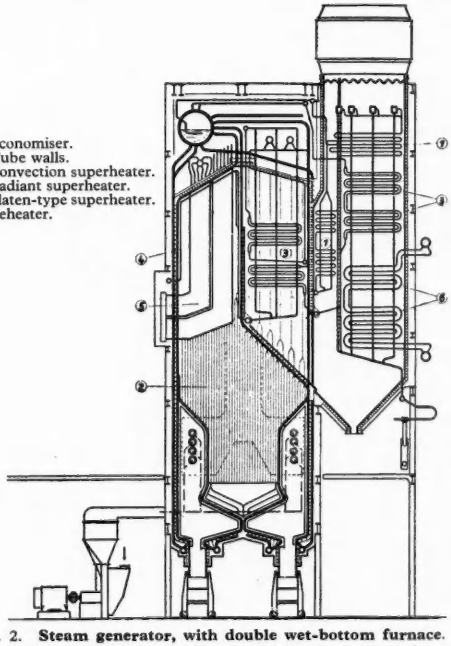

Fig. 2. **Steam generator, with double wet-bottom furnace.**
Evaporation 480,000 lb/hr ; 1940 psi ; 985° F ; single-stage reheat.

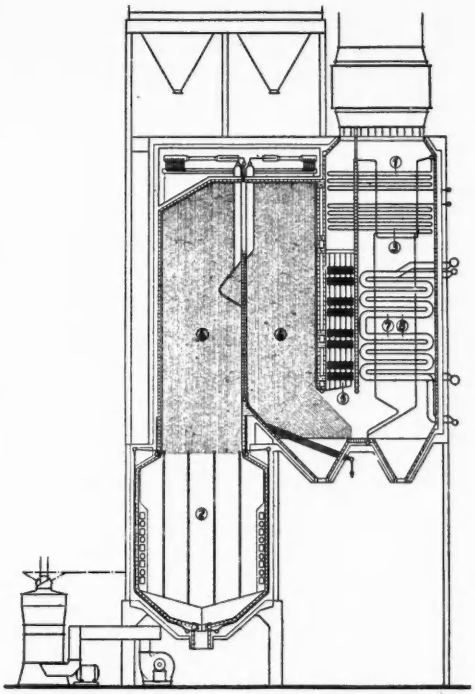

Fig. 3. **Steam generator with forced circulation.** Evaporation 560,000 lb/hr; 4400 psi; Boiler steam-outlet temperature 1120° F; two-stage reheat.
(1) Economiser.
(2) Combustion chamber.
(3) Demineralisation zone.
(4) Superheater, nominal pressure 4270 psi, 1st stage.
(5) *Ditto*, 2nd stage.
(6) *Ditto*, 3rd stage.
(7) Reheater, nominal pressure 1780 psi.
(8) Reheater, nominal pressure 450 psi.

dynamic calculations. Considering that, at prevalent temperatures, an extra 30° F may mean a difference in the mechanical properties of the order of 50%, every effort must be made to limit excess temperature.

In an analysis of this tube-to-steam temperature difference, three major components must be considered. Of these, the most important is the temperature difference between the inside and the outside of the tube. This depends on economic considerations which determine optimum tube diameter and this, in turn, establishes the wall thickness, the tube surface, and rate of heat transfer. Improvement of heat transfer by increasing steam velocity is limited by considerations of friction losses. It can thus be seen that this component is not amenable to reduction.

The second component, i.e., *local* temperature differences, caused by uneven heating and steam flow, will respond to careful arrangement of cross-overs, mixing-headers, and means of producing even steam flow.

The third component covers the delay in response to temperature control. The greater the mass of metal in the superheater to be controlled, and the larger its steam content, the longer is the time lag, even if the control device reacts immediately to the impulse. It follows that the desuperheater should be located as near as possible to the steam outlet. A second desuperheater located in an intermediate position would take care of starting-up difficulties and of peak loads.

There still remains a small amount of inertia which can, however, be compensated by a secondary signal superimposed upon the temperature control-device. This secondary signal could be suitably derived from such points as the steam metering device, or, in the case of units using automatic combustion-control, from the air-flow measuring point, and would induce the temperature control to take preventive action.

Materials

Application of higher temperatures has been made possible by the development and long-term testing of materials suitable for continuous boiler operation under exacting conditions. At present, scantlings for pressure parts are based upon strength values existing after 100,000 working hours and upon the permanent elongation caused by operating conditions during that period. Up to now, austenitic steels have proved themselves adequate. Reliable welding techniques have been developed for joining austenitic steels or combinations with ferritic steels. Boiler development is bound up with welding development; practically all connections, including mountings, on very high-pressure boilers are welded, and attempts are being made to replace the expanding of tubes into the drums by welding.

Special problems of raising steam at super-pressures

For the purpose of collecting operational data in the super-pressure range, a pilot-plant, illustrated in Fig. 4, has been installed. It consists of a vertical closed loop and operates at about 4300 psi. The riser is heated to 1110° F inside a flue, which is about 25 ft high, whilst the downcomer is being cooled at the top end and the water re-enters the riser at 570° F. Thus, a natural circulation is created. A small, steam pressure-controlled piston-pump tops up feedwater when necessary, and steam temperature is maintained by burner control.

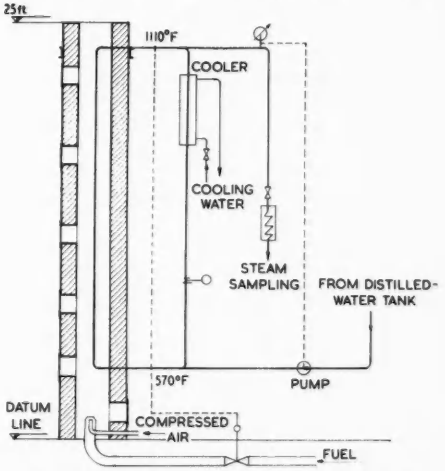

Fig. 4. **Pilot-plant for 4300 psi and 1110° F.**

Steam samples can be taken continuously from the downcomer and numerous temperature measuring points are inserted into the riser. Practical answers are expected to questions concerning the actual process of evaporation, the suitability of materials, and the demineralisation-process and its temperature range and chemical reactions with tube materials. Calibrated tubes of varying thickness can be inserted, and results under continuous operating conditions compared with values found by destructive test methods. Different types of tube connections between austenitic and ferritic materials will enable conclusions to be drawn regarding full-scale applications.

U.S.A.

A Technique for Corrosion Testing in Liquid Lead

By J. V. CATHCART and W. D. MANLY. (From *Corrosion*, Vol. 10, No. 12, December 1954, pp. 432-434, 6 illustrations.)

IT has long been evident that the thermal properties of liquid metals should make them excellent heat-transfer media. Work in this field was pioneered by the development of the mercury boiler.[1] In recent years, the use of liquid metals as coolants in nuclear reactors, as well as in a variety of industrial applications, has received serious consideration.[2]

The increasing interest in liquid metals has focused attention on the accompanying corrosion problems and the techniques for corrosion testing in these materials. One of the most serious types of corrosion encountered has been that of mass transfer, i.e., the transport of container material from the hot to the cold zone of the system under consideration. Any testing programme designed to determine the most suitable container material to be used with a given liquid metal must take mass transfer into account; therefore, the testing apparatus must provide for the circulation of the liquid metal through a temperature gradient. Simplicity of design and operation is also important, because frequently many experiments are required before a choice of the proper container material can be made.

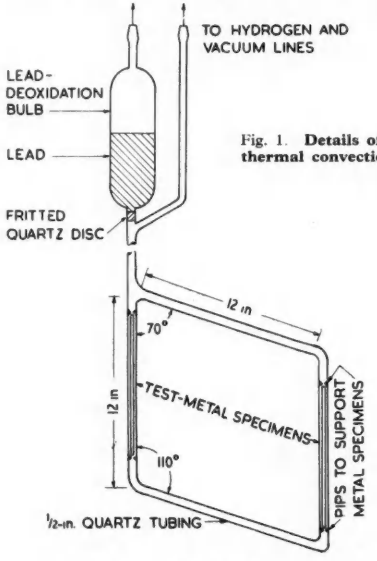

Fig. 1. **Details of quartz thermal convection loop.**

The apparatus described in this article meets the foregoing specifications for tests involving liquid lead. It is a modification of the thermal-convection loops which were used extensively in the development of the mercury boiler.[1, 3] As shown in Fig. 1, the loop was made of ½-in. quartz tubing. Six-inch tubes of the test metal were mounted in the hot and cold legs of the loop, where they were held in place by indentations or pips in the quartz tubing. Attached above the loop and separated from it by a fritted quartz filter disc was a large quartz bulb, 80 mm in diameter. This bulb was used for the deoxidation of the lead prior to the loading of the loop. Two side arms, one above the deoxidation bulb and the other below the fritted disc, were attached to hydrogen and vacuum lines through a suitable system of stopcocks. Eight-inch split-core clamshell heaters were used to heat the loop, one such heater being suspended by stainless-steel clamps around each arm of the loop. A fifth heater was placed around the tube connecting the loop and the deoxidation bulb. The bulb was heated with electric tape heaters. Temperatures were measured by thermocouples spaced around the loop and tied in place with wire.

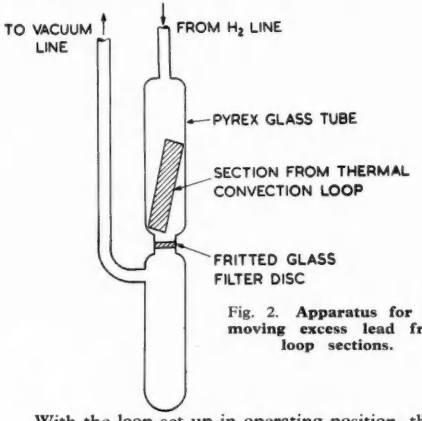

Fig. 2. **Apparatus for removing excess lead from loop sections.**

With the loop set up in operating position, the lead in the deoxidation bulb was heated to 425° C, and hydrogen was allowed to pass through the fritted quartz disc and to bubble up through the lead. During the deoxidation process, which lasted for about 48 hours, surface-tension forces prevented the liquid lead from passing through the fritted disc. When the deoxidation was complete, the loop was evacuated and hydrogen was admitted into the deoxidation bulb; this caused the lead to flow through the fritted disc and into the loop. The loop having been pre-heated to operation temperature, the heaters on the cold arms were turned off and circulation of the lead was maintained by the thermal-convection currents thus induced.

The above procedure assured that the oxide content of the lead was reduced to a minimum and, in addition, prevented reoxidation of the lead during the loading of the loop. The technique also offered several other advantages. Each loop required only a small amount of metal tubing, a factor of some importance in cases where the test metal is expensive or in short supply. No bending or welding of the test metal was required; consequently, such hard-to-fabricate materials as metallic chromium or molybdenum could be studied without difficulty. The fact that the test specimens were completely enclosed in the quartz tubing of the loop eliminated any necessity for protecting the outer surface of the test metal from air oxidation; thus, the testing of metals such as molybdenum or columbium was greatly facilitated. The detection of leaks in a quartz loop prior to loading was also much easier than in a conventional metallic loop.

The relative resistance to mass transfer of one test metal as compared with another was obtained by measuring the time required for a sufficient quantity of mass-transferred material to build up in the cold leg of the loop and cause a stoppage in the circulation of the liquid lead. This process is referred to as "plugging." The plugs were always a porous matte of fine dendritic needles and plates. A wide variety of metals and alloys has been tested by this method. A wide range of plugging times was observed. The loops containing nickel plugged in less than two hours, while no mass transfer at all occurred in over 600 hours in the columbium loop, thus proving that the technique

possessed a reasonable degree of sensitivity.

The extent of mass transfer could be estimated very roughly during the course of an experiment from measurements of the cold-leg temperature. As a plug built up, the circulation of the lead was slowed, thus producing a decrease in the cold-leg temperature. Such behaviour was found to be typical of plug formation. When plugging was complete, a sharp drop in the cold-leg temperature occurred and the lead in the cold leg solidified.

After a test had been completed, the loop was cut into short sections, and each section was placed in a tube, such as that shown in Fig. 2. These tubes were

(*Concluded on page* 85).

AUSTRIA

Investigation into the Mechanics of Cavitation

By G. ZIEGLER. (From *Maschinenbau und Wärmewirtschaft*, Vol. 9, No. 12, December 1954, pp. 343-349, 15 illustrations.)

ALTHOUGH published literature[1-6] contains various references which appear to indicate the possibility of cavitation starting at pressures differing from the vapour pressure, there have been no systematic measurements available which would determine these pressures as functions of the flow velocity and air content.

To clarify this question, cavitation tests were carried out by the author, using a test installation of a size and type which would give data in the range of flow conditions occurring in flow-type machines.

The test installation was designed as a closed cycle and operated by means of a medium-pressure pump of 60 hp. The smallest cross-section of the measuring section was approximately 20 cm². By variation of the air pressure in a pressure vessel set up outside the circuit and connected to it only by a narrow pipe, it was possible to alter the pressure level of the cycle without having, at high pressures, appreciable quantities of air dissolved in the water. This arrangement enabled cavitation to be produced in the test section at velocities between 6 and 30 m/sec. When the cycle was operated at high pressures, the water was enriched with air (up to about 8% by volume) by means of an air injector. On the other hand, when the installation ran at a low pressure level for an extensive period with considerable cavitation in the measuring section, it was possible by means of a de-aerator, connected at the end of the test section, to reduce the air content of the water used in the tests down to about 0·8% by volume.

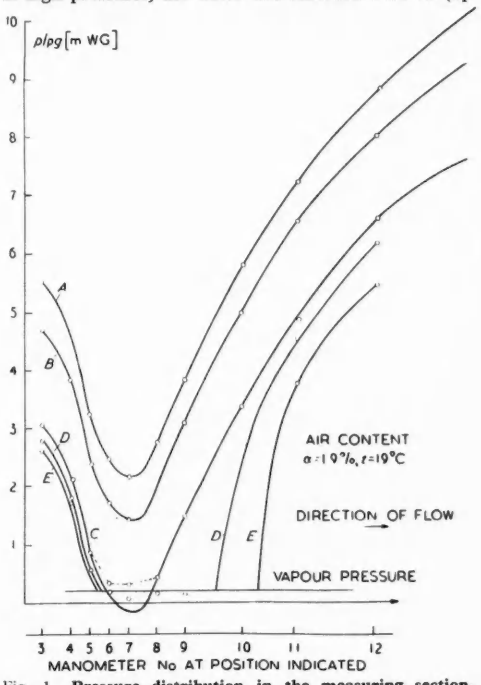

Fig. 1. **Pressure distribution in the measuring section, with various pressure levels at constant flow volume.**

In Fig. 1, curve *A* indicates the pressure distribution measured in a venturi tube. By gradual reduction of the pressure level of the cycle, with constant flow volume, the pressure distributions of curves *B*, *C*, *D*, and *E* were obtained. Curve *C* corresponds to the condition where cavitation sets in. When vapour pressure is reached, the water in the measuring ducts begins to boil, so that the pressures below the vapour pressure cannot be indicated. The manometers in the narrowest cross-section gave the values plotted in Fig. 1 with a dotted line.

When the first few minute vapour bubbles appear, and disappear again after a brief existence, the continuity is hardly disturbed. The true pressure distribution in the corresponding region may therefore be expected to have the form followed by curves *A* and *B*, and no longer that indicated by the measured values. Curve *C* was completed in this manner over the region of the narrowest cross-section and its lowest point thus obtained indicates the minimum-pressure position, which corresponds to incipient cavitation. The water, therefore, has been subjected to tension stresses before even reaching the condition of incipient cavitation.

With a further reduction in the pressure level, curves *D* and *E* were obtained, corresponding to more advanced stages of cavitation. The continuity in these cases is already disturbed by considerable vapour-bubble formation. Moreover, pressure losses occur, so that with fully developed cavitation it does not appear justified to complete the curves in the negative-pressure region.

In the main test, the negative-pressure peaks were determined, these occurring in the narrowest cross-section when the first vapour bubbles appeared. The pressure p_c occurring at a cavitation position c was evaluated from the measured pressure p_b, occurring at the beginning of cavitation in a channel cross-section b situated ahead of the narrowest section, and from the pressure difference $(p_b' - p_c')$ situated at a higher pressure level (above the vapour pressure) but measured at the same velocity in the cycle of the water, using the relation

$$p_c = p_b - (p_b' - p_c').$$

It was considered justifiable to employ this relation, in view of the small extent of bubble formation during incipient cavitation. To eliminate the disturbances originating from the pressure tap, when determining the beginning of cavitation, the manometers were connected to the narrowest position only after determining the flow volume and the pressure level at the time of incipient cavitation.

The influence of the finite diameter of the pressure

taps was also found to have a noticeable effect[7] during the measurement of static pressures. With local dynamic pressures, reaching values of about 60 metres of equivalent water gauge, an error of 1% in the indication of the measuring section corresponded to about 0·6 metres water gauge—an appreciable amount, in view of the required accuracy for static pressure determination. The necessary corrections were determined experimentally after completing the main series of cavitation measurements. These corrections were found to have values considerably higher than those obtained in the tests made by Miyadzu[7], probably because the velocity gradients near the wall were much higher than in his tests.

The condition of incipient cavitation was determined optically, acoustically, and (to eliminate subjective errors in observation) by means of a highly sensitive electrical vibration-measuring device.

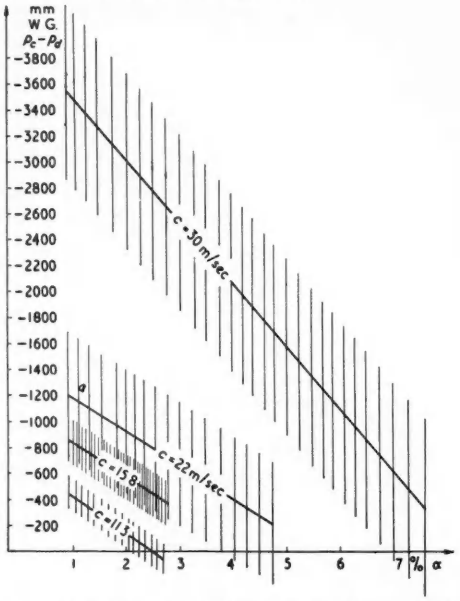

Fig. 2. **Negative pressures below vapour pressure in the narrowest cross-section of flow, with various flow velocities and air contents.**

The results of the pressure determinations at incipient cavitation are shown in Fig. 2. The base-line gives the values of the air content α of the water (percent by volume) related to 0° C and 760 mm Hg. The ordinates indicate values of the pressure difference $p_c - p_d$, where p_d is the vapour pressure corresponding to the water temperature. The values of $p_c - p_d$ are all negative. Thus, Fig. 2 does not indicate "tension stresses" p_s but "negative pressures below vapour pressure." As, during the tests, the water temperature fluctuated between 18° and 26° C, it appeared more advisable to relate the minimum pressures to the vapour pressure. A check test gave no evidence that a change in pressure due to temperature, during incipient cavitation, would exceed the range of variation of the vapour pressure with temperature.

The vertical lines in Fig. 2 denote the mean range of scatter for all the quantities occurring in the final results and can be regarded as a measure of the reliability of the measurements. They are primarily due to the scatter obtained in taking pressure measurements with incipient cavitation on different days under otherwise identical conditions, and also to the uncertainty of the corrections for measured pressures in order to take account of the effect of the size of the measuring channels.

The limit on the left-hand side of the straight lines $p_c - p_d = f(\alpha)$ is due to the performance limitations of the de-aerator and the vacuum pump. There was also a limit on the high side for the α-values within which tests could be carried out.

In tests with water having a low air content, the first appearance of cavitation, as pressure decreased, was a thin layer of bubbles on the wall. All the test results indicate that, when the water enters the negative-pressure region, there first occurs a separation of air bubbles, which serve as nuclei of evaporation, the water evaporating into these on reaching vapour pressure. Thus, the formation of vapour bubbles was preceded by de-aeration. It was not possible to identify optically the point at which evaporation into the air nuclei commenced.

When using water with a high air content, it was found that another form of de-aeration took place. During the falling-off of pressure, the water in the nozzle suddenly took on a dull and milky appearance. This de-aeration took place not only near the walls but over the entire cross-section, without at first any occurrence of cavitation. The latter appeared only after a further reduction in pressure, in the form of a veil-like layer of vapour bubbles, clearly distinguishable from the air separation. The tests with gradually increased air content had to be discontinued when this preliminary air separation began to occur, since under its influence the air content of the water in the test was altered to such an extent that corresponding values of pressure and air content were no longer obtainable.

On the basis of these observations, Fig. 2 may be interpreted as follows:—Where the air content was so great that air separation occurred before incipient

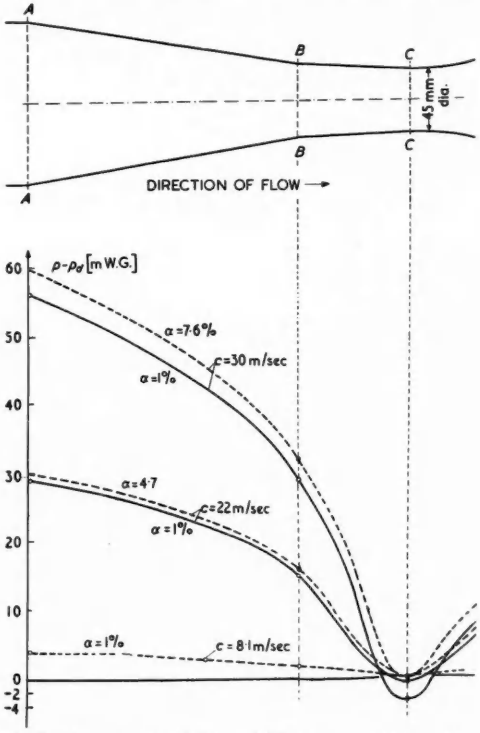

Fig. 3. **Pressure distribution within the measuring section.**

cavitation at the velocity investigated (end points at the right-hand end of the line $p_c - p_d = f(\alpha)$) sufficient nuclei were present for evaporation. At all velocities, cavitation began when vapour pressure was reached. With lower air contents, on the other hand, these nuclei were lacking or were not sufficiently numerous, so that it was necessary to go below the vapour pressure in order to obtain any bubble formation. The amount of negative pressure required below the vapour pressure increases with decreasing air content, i.e., with a decreasing number of evaporation nuclei. It also increases with higher flow velocities of the water, i.e., when the time available for the formation of evaporation nuclei is reduced. The pressure distributions which lead to the incipient cavitation are schematically represented in Fig. 3, in which the pressure distribution between measuring points is an assumed curve.

In addition to the measurements described above, long-exposure photographic recordings were taken with the arrangement shown in Fig. 4. Some typical conditions are illustrated in Fig. 5. Fig. 5a refers to water with a high air content ($\alpha = 3\%$) at a velocity $c = 18$ m/sec in the narrowest cross-section. The flow is from left to right. The positions at which cavitation begins and those of the breakdown of bubbles occupy the entire cross-section of the channel, the phenomenon occurring at one cross-section position in each case. The slight disturbance indicated by a somewhat separate stream at the bottom of the picture is caused by a small roughness to the left. A soft rustling noise was audible.

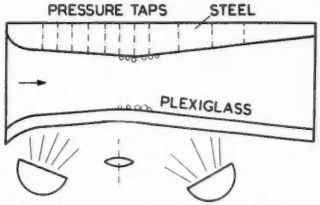

Fig. 4. **Schematic arrangement for long-exposure photographic recordings.**

When the velocity was reduced to 12 m/sec (Fig. 5b), an even more uniform picture was obtained. The irregularity due to the slight disturbance disappeared and the noise became even weaker. However, in contrast with conditions at higher velocity, a marked air separation occurred in this case, owing to the low static pressure in the cycle, although the air content was the same as previously. After the position of bubble breakdown, there was a further stream of entrained air bubbles, also clearly noticeable.

With the air content reduced to $\alpha = 0{\cdot}8\%$, at $c = 18$ m/sec (Fig. 5c), the vapour trails occurred behind wall roughnesses only and not on the smooth wall. The separate streams of vapour bubbles originated behind the roughnesses caused by two pressure taps (of only 0·4 mm diameter) and slight roughnesses in the throat of the rectangular-section channel. A hard, grating noise was produced. At $c = 12$ m/sec with the same air content, the trails became slightly wider, but again originated only at disturbance points.

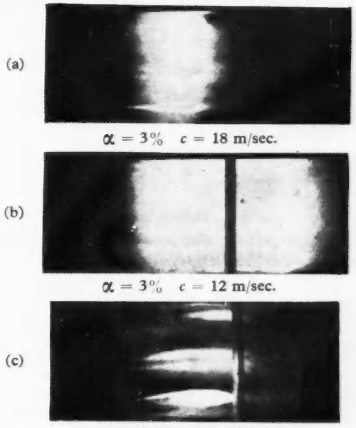

Fig. 5. **Long-exposure photographs of flow phenomena, using arrangement shown in Fig. 4.**

Other photographs were recorded by means of a similar arrangement to that shown in Fig. 4, except that illumination was obtained from spark discharges at 40,000 V, with a duration of about 1 microsecond.

These photographs, two of which are reproduced in Fig. 6, give further details of the flow phenomena. The two vertical dotted lines in Fig. 6a denote the extent of the narrowest cross-section of the nozzle. With a high air content ($\alpha = 3\%$) and low velocity, cavitation developed uniformly over the entire cross-section. At the higher velocity ($c = 22$ m/sec), with the same air content, the vapour bubbles are more irregularly distributed, as shown. Behind the breakdown position, a subsidiary flow of minute air bubbles is noticeable.

Fig. 6b shows a highly magnified view of the bubble formation in the vicinity of the wall. The small bubbles are seen flowing in from the left; these are probably air bubbles. The deformation of the bubbles can then be followed, as it occurs under the influence of the very steep velocity profile in the neighbourhood of the wall.

The Dynamics of Bubble Formation

The detrimental effects of cavitation in hydraulic installations have stimulated interest in the mechanics of cavitation, with the object of clarifying such questions as the origin, growth, and breakdown of the bubbles. Nevertheless, the explanations provided are not yet

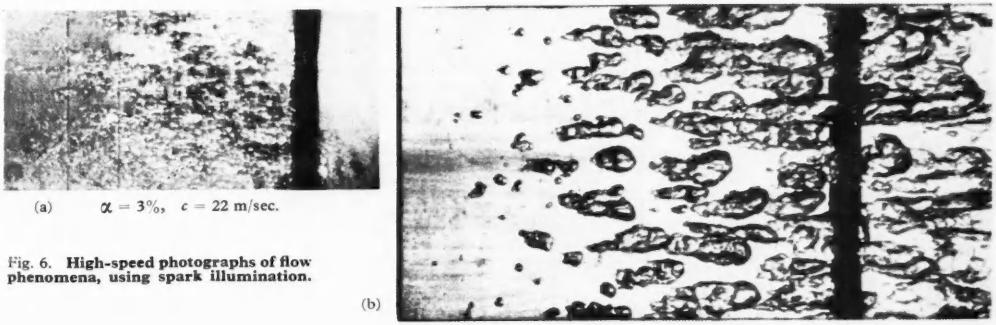

Fig. 6. **High-speed photographs of flow phenomena, using spark illumination.**

fully satisfactory, particularly with regard to the breakdown process. Since 1917, when Lord Rayleigh[8] first published his differential equation for the equilibrium between pressure and mass forces, there has not been an appreciable increase in the knowledge of this subject. A remarkable step forward, however, was achieved by Knapp[6] in 1952, who showed by means of high-speed photographic recordings that the breakdown of bubbles occurs approximately as described by Rayleigh's equations.

This justifies the conclusion that mechanical forces are the main factors, whereas surface tension and thermal processes are of secondary importance. The differential equation is as follows:—

$$-\Delta p = \rho \left[R \frac{d^2R}{dt^2} + \frac{3}{2}\left(\frac{dR}{dt}\right)^2 \right]$$

where Δp = pressure difference between the water pressure in the outer vicinity of the bubble (theoretically, at " infinite distance " from it) and the vapour pressure prevailing inside the bubble, ρ = mass density of the fluid, and R = radius of the vapour bubble.

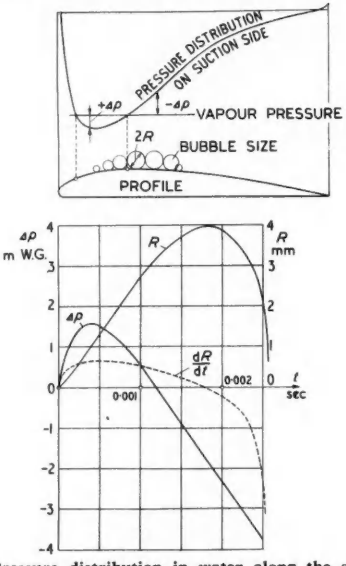

Fig. 7. **Pressure distribution in water along the suction side of the hydrofoil shown in the upper portion of the figure, and change in bubble radius while passing through the pressure field, plotted against time.**

By means of this equation, for a given pressure distribution and with the assumption that a nucleus for evaporation exists, the growth and breakdown of individual spherically-shaped bubbles can be calculated. Fig. 7 shows a given pressure distribution on the suction side of a hydrofoil. The velocity distribution is known, and hence the water pressure is also known in the outer vicinity of the bubble, at any instant. The difference in pressure between the interior of the bubble and its outer vicinity first has the tendency to increase the bubble diameter. The water surrounding the bubble moves away from the bubble centre, and, owing to its inertia, retains this motion thereafter, even when the outer pressure is again equal to ($\Delta p = 0$) or even larger than ($\Delta p < 0$) the pressure inside the bubble. After the reversal of motion, which occurs when the bubble attains its maximum diameter, the bubble contracts at an increasing rate. These considerations apply to the case where the pressure field around the bubbles is not appreciably disturbed, i.e., where there are still a few small vapour bubbles present. If the vapour bubbles occupy a larger space, a considerable rise in pressure occurs during and as a consequence of their breakdown; this pressure rise is superimposed upon the field of the undisturbed flow. This case has been investigated by Ackeret[9]. The correspondence between the positions of the negative-pressure peak, the maximum bubble diameter, and the position of bubble breakdown (Fig. 7), as well as the fact that agreement was obtained in another example between calculation and experiment, seems to justify the treatment in this example.

It still remains to be shown whether there is agreement to some extent between the results obtained and those previously published. Referring to the short periods of motion of the water particles, Ackeret[9] pointed out the possibility of the appearance of tension stresses. Crump[5] reported on the measurement of tension stresses corresponding to a negative head of about 20 metres of water column, obtained, however, under very unstable conditions. Finally, Knapp[6] measured tension stresses at various velocities and for various sizes of models. His results agree qualitatively with those presented in this article. Knapp used the air content as a parameter, without specifying its numerical value, and stated that the air content was a basic but not sufficiently characteristic parameter for liquids. He based this assertion on a test carried out by Harvey, McElroy, and Whiteley[10], in which water samples were investigated as regards tension strength, then subjected to a pressure of 700 atm, and again tested for tension strength. The water after the pressure treatment, with the same content of air dissolved, resisted considerably higher tension stresses. From this Knapp draws the conclusion that it is not the air content but the quantity and size of the minute air bubbles in the liquid which characterise its cavitation behaviour. However, the cavitation coefficients determined by various investigators on the basis of the air content do not show an appreciable amount of scatter, so that, for practical purposes, characterising a liquid by its air content is justified. All measurements of the influence of air content showed increased danger of cavitation with increased air content.

On the basis of his tests and without taking air content into account, Knapp derived a relation for the variation of the cavitation parameter σ as a function of the pressure head and model size. This formula leads one to expect marked variations of σ for the case of a turbine with small heads, but only very minor variations when operating with larger heads. This theoretical consideration is in agreement, qualitatively, with tests on turbine installations[11, 12]. It should be noted in this connection that, in measurements with very small heads, the variations of the cavitation pressure are not the only factor, since the air separation in the draft tube also has an influence both on the flow and the measurements. If the diagram in Fig. 2 were modified so as to obtain a representation of $\sigma = f(c, \alpha)$, the variations of σ thus obtained would be so small that it would be inappropriate to try to derive from these a dependence of the cavitation parameter σ on the head and model size, taking account of the air content.

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The Influence of Surface Tension on the Formation of Graphite in Cast Iron

By K. GRÜTTER and B. MARINCEK. (From *Archiv für das Eisenhüttenwesen*, Vol. 25, No. 9/10, September/October, 1954, pp. 447-453, 16 illustrations.)

SINCE spheroidal graphite cast iron was first produced on a technical scale a few years ago, considerable research has been carried out on the mechanism of graphite formation, but there is still a scarcity of information on the essence of the process itself, i.e., why it is possible for the graphite to be formed in a spheroidal rather than a flake structure. A number of investigators have tried to explain the formation of spheroids by the assumption that the crystallisation processes were influenced by the surface tension of the cast iron, though they have not undertaken the relevant measurements. These have been carried out and are reported by the present authors.

The results of these measurements show that, while the surface tension of ordinary pig iron and cast iron ranges from 800 to 1100 dynes/cm, that of magnesium-treated spheroidal graphite cast iron lies between 1300 and 1400 dynes/cm, i.e., from 40 to 50% higher than the surface tension of the basic material produced in the cupola furnace. A close correlation between surface tension and sulphur content was also established.

These results seem to confirm the assumption that surface tension may exert an influence on the form of the graphite segregated in the cast iron, and additional experiments, with a view to ascertaining its influence on the crystallisation of the graphite, were indicated.

Melting was entirely carried out in a 30-kW, 10,000 cps, medium-frequency furnace, with lined crucibles. Samples of 2 kg were melted down as rapidly as possible and, to prevent excessive oxidation, argon was introduced into the open crucible during melting and while the melt was held at temperature. The bath having been adjusted to the desired temperature immediately after complete melting, the surface tension was measured by a capillary depression process described elsewhere[1]. Surface tension was measured with a calibrated Pt/PtRh thermocouple and automatically recorded, the temperature ranging from 1320° to 1340° C in some experiments, and from 1390° to 1410° C in others.

The experiments were carried out on the following series of melts:—

Series A:—A specimen of cast iron melted in a cupola furnace was cast in sand. The cast iron remaining in the ladle was treated with 0·5% pure magnesium[2], then—before the ferro-silicon inoculant was added—a further specimen was cast in sand. A third specimen was taken after 0·5% of 90% ferro-silicon had been added (*Series AI*). A similar set of specimens was taken from a second series of melt from the cupola furnace, but here 0·4% pure magnesium was added (*Series AII*).

The specimens were melted down in the medium-frequency furnace under the above-mentioned conditions, the surface tension was measured, and the cast iron examined with regard to its graphite structure.

Series B:—To determine the influence exerted on surface tension, and hence on the development of the graphite structure, by the length of time for which a cast-iron melt is held at a certain temperature, the magnesium-treated and inoculated samples of Series AI and AII tests were held for long periods at about 1300° C (*Series BI and BII*) or about 1400° C (*Series BIh* from the iron of Series AI). At certain intervals, measured from the completion of melting, surface tension was measured and the graphite structure of the iron examined. After each measurement, a specimen of 40 to 50 grams was taken from each melt for an examination of its chemical composition.

Series C:—Ordinary haematite (2 kg) was melted down in the medium-frequency furnace and treated with 0·4% and 0·7% pure magnesium[2]. Surface tension was measured immediately after the introduction of the magnesium and before inoculation with ferro-silicon, and again a few minutes after the addition of magnesium.

Series D:—In view of the authors' earlier findings that there was a close correlation between surface tension and sulphur content[1], the influence of surface tension on graphite structure in the presence of iron sulphide was investigated. A synthetic cast iron was produced from Armco iron, graphite, and ferro-silicon, the iron sulphide was added in increasing amounts, and the surface tension was measured. The synthetic cast iron was further treated with magnesium, which was also added in increasing amounts, surface tension being measured after each addition. The specimens obtained were subjected to metallographic examination.

RESULTS OF EXPERIMENTS

The basic material used in the Series AI and AII tests is characterised by long, coarse flakes. Undercooled graphite is almost entirely absent (Fig. 1). If sufficient magnesium is introduced into this cast iron, a spheroidal graphite structure results, with the graphite spheroids embedded in a ledeburite-base material (Fig. 2). In Series AII tests, where 0·4%, instead of 0·5%, pure magnesium had been introduced, the surface tension, amounting to 1308 dynes/cm, no longer suffices to form the spheroids, and only compact rosettes were present, apart from undercooled graphite (Fig. 3). However, the magnesium-treated basic material of this series, i.e., before the specimens were melted down, still contained spheroids. The effect of the duration for which these melts were held at a definite temperature (Series BI, BIh and BII) is highly interesting. In Series BIh tests (holding temperature 1390° to 1410° C), 9 minutes after the commencement of fusion the graphite is entirely in the form of nodules; after a further 7 minutes, they disappear and are replaced by rosettes and undercooled graphite (Fig. 4), and surface tension drops by 100 dynes/cm. Continued holding reduces the surface tension still further, though no significant change in structure occurs, except that the initially very compact rosettes assume a more laminar character (Fig. 5). After 28 minutes from the start of melting, the surface tension drops to 1230 dynes/cm, and the rosettes are replaced by flakes. A small amount of undercooled graphite is still present. The same process can be observed in Series BI tests (holding temperature approximately 1310° to 1340° C), but the spheroids are still present after 14 minutes, though tending to disintegrate (Fig. 6). In Series BIh tests, owing to the higher holding temperature, the spheroids disappear more rapidly than in the Series BI tests. Diminution of the surface tension is accompanied by a fall in the magnesium content of the melts, this being more rapid at higher holding temperatures. Spheroids are absent when the magnesium content is still at approximately 0·07%.

The diminution of the surface tension, and hence the disappearance of the spheroids, may be explained as follows:—The magnesium content of the melt decreases as the holding time increases. At the same time, if a melt of spheroidal graphite cast iron is held at an elevated temperature for a long time, a "resulphurisation" takes place, which may attain 50% of the original sulphur content[3]. The cupola-furnace iron of the Series AI tests had a sulphur content of 0·086%.

Fig. 1. **Structure of Specimen 1 (unetched) of Series AI tests. (× 90)**

Fig. 2. **Structure of Specimen 3 (unetched) of Series AI tests. (× 90)**

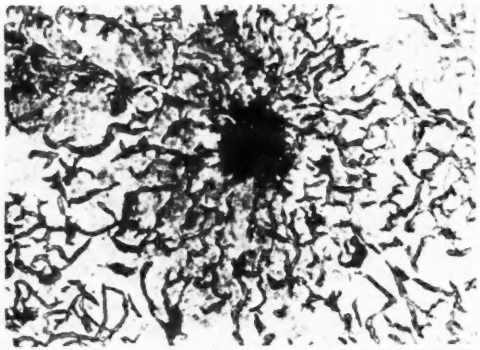


Fig. 3. **Structure of Specimen 3 (nital-etched) of Series AII tests. (× 360)**

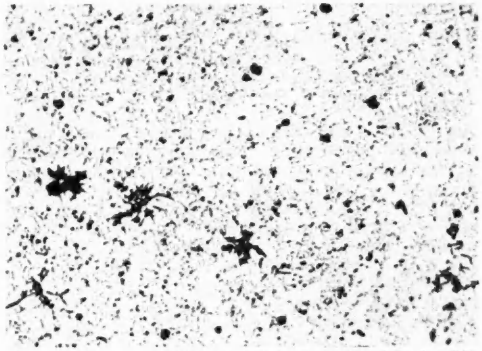


Fig. 4. **Structure of Specimen 2 (unetched) of Series BIh tests. (× 90)**

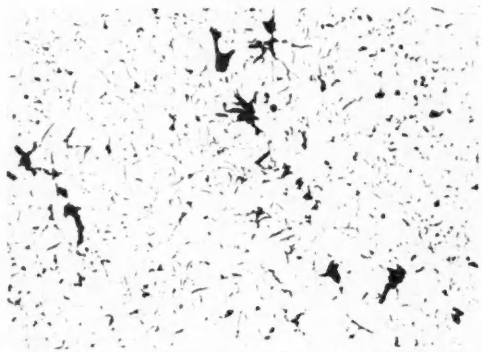


Fig. 5. **Structure of Specimen 3 (unetched) of Series BIh tests. (× 90)**

Fig. 6. **Structure of Specimen 2 (nital-etched) of Series BI tests. (× 360)**

Immediately after the magnesium had been added, it dropped to 0·002%, but, when the material had been re-melted and held at slightly over 1400° C for 30 minutes, the sulphur content was 0·02%, so that the re-sulphurisation amounted to nearly 20% of the initial content.

This re-sulphurisation reduces surface tension, as even a slight sulphur content is sufficient to produce this effect. Furthermore, despite the introduction of argon, atmospheric oxygen may nevertheless exert some slight influence, and this cannot be ignored if it is assumed that oxygen affects surface tension in the same way as sulphur. Hence, the diminution of surface tension in a melt with spheroidal graphite after a long holding period, and the disappearance of the spheroids, may be ascribed to re-sulphurisation coupled with oxidation. The decrease of magnesium content caused by oxidation or evaporation also counteracts the formation of graphite nodules.

In the Series BII tests, the holding temperature was slightly above 1300° C. At the first measurement of surface tension, 9 minutes after the start of melting, there were already no spheroids, only rosettes and a considerable amount of undercooled graphite; subsequently, the rosettes were replaced by small flakes.

In the Series C tests, the haematite used had a surface tension of 950 dynes/cm at 1330° C; 2 minutes after the addition of 0·4% Mg was completed, this figure was 1335 dynes/cm. Graphite nodules were present in a white base melt; 1·5 minutes after 0·7%

Mg had been added, the surface tension was 1375 dynes/cm, and the solidified melt was entirely white (Fig. 7); 5 minutes after the addition of the Mg, the white base melt still contained spheroids, though they distinctly tended to disintegrate. At 1340° C, the surface tension was still 1338 dynes/cm. After a total of 12 minutes, the spheroids disappeared, leaving only undercooled graphite and spheroidal rosettes (Fig. 8).

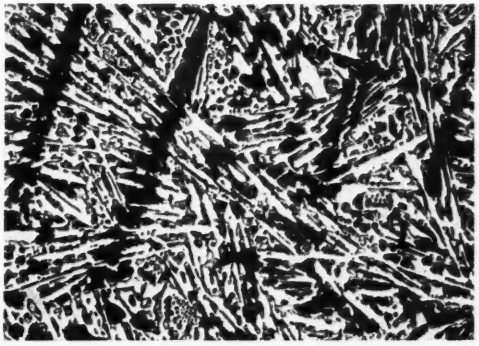


Fig. 7. **Structure of Specimen 3 (nital-etched) of Series C tests. (× 180)**

In the Series D tests, surface tension changed within the range of approximately 600 to 1400 dynes/cm, when the synthetic iron was treated with iron sulphide or pure magnesium. The graphite structure was examined in a specimen cast in sand. Within the surface-tension range of 1300 to 1400 dynes/cm, the base melt solidified white, and the graphite nodules were surrounded by austenite. This range relates to specimens 1 and 2 of Series D, with 0·7% and 0·4% Mg respectively. The specimen with 0·1% Mg shows a preponderantly laminar structure, but there are also zones where graphite is present only in nodules of small diameter. In the transition zone between spheroids and flakes, the form of the graphite accords with that of undercooled graphite. This transition is clearly shown in Fig. 9. The melt with the next lowest surface tension had been sulphurised to 0·090%. The graphite precipitate consists of thick, coarse flakes, the base melt of the specimen being pearlitic, as in the previous case. The specimens with still higher sulphur contents, i.e. lower surface tensions, are characterised by the fact that the coarse flakes become more compact. Again, there is a white base melt, but it is far more coarse-grained and irregular than that of the magnesium-treated specimens. In these, a distinctly eutectic mixed structure of primary dendrites was formed in a laminar base melt, as shown in Fig. 7.

Conclusions

The results of these experiments show that a certain form of the graphite in cast iron is linked with a certain surface tension. Thus, within the range of 1300 to 1400 dynes/cm, graphite nodules are present, followed at lower levels by undercooled graphite with individual rosettes. While in cast iron with spheroidal graphite the base melt solidifies white, in this case it

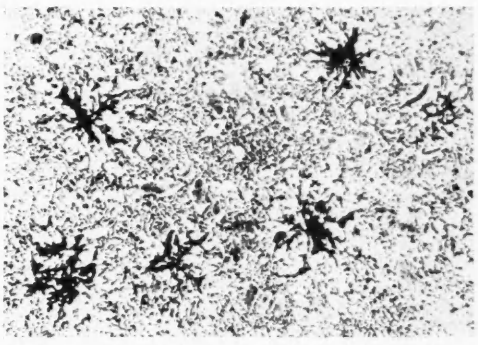


Fig. 8. **Structure of Specimen 5 (nital-etched) of Series C tests. (× 180)**

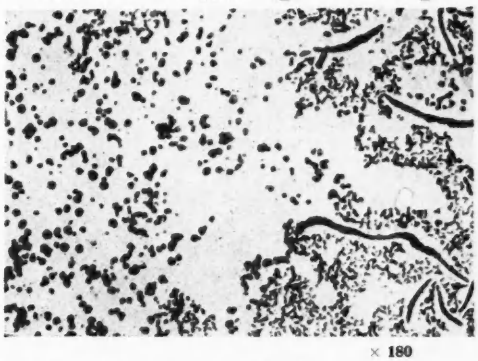
is pearlitic-ferritic. Within the range of 800 to 1100 dynes/cm, the graphite is mainly present in long flakes. This is ordinary pig iron and cast iron with a pearlitic base melt. Below 800 dynes/cm, the flakes again become more compact and may once more become spheroidal, while the base melt is again white, owing to the carbide-stabilising effect of the high sulphur content.

The disappearance of the nodules from a melt of spheroidal graphite cast iron held in the ladle for a long time may be ascribed to re-sulphurisation and oxidation, and to concurrent diminution of surface tension.

It follows from these results that the form of the graphite can be considerably influenced by surface tension. If surface tension is low, there is practically no resistance to the crystallisation of the graphite and flakes can form; if it is higher, crystal growth is resisted and long flakes cannot form. As surface tension increases, the flakes become narrower and shorter, until finally, at a very high surface tension, this resistance is so great that the graphite formed possesses the minimum surface area. Rosettes, undercooled graphite, and finally spheroids are produced. Hence, the production of spheroidal graphite cast iron can be promoted by measures designed to increase surface tension.

These conclusions apply only to the cast iron melts

× 90



× 180

Fig. 9. **Structure of Specimen 3 (nital-etched) of Series D tests in the annealed state.**

used in the present experiments, and only further investigations can determine whether they apply generally. Unfortunately, this work does not establish the extent to which surface tension exerts a direct influence on the form of the graphite. At the higher levels, surface tension is also a function of the magnesium content, and it is quite possible that the spheroidal form of the graphite is influenced by a hitherto unknown property of the magnesium addition. On the other hand, the graphite form may be determined by the surface tension of the surrounding phase at the time of graphite growth, whereas the measurements recorded here relate to the liquid phase and hence to temperatures higher by several hundred degrees, so that these figures can be only approximate as regards graphite formation.

In any case, it is clear from the results obtained that, under certain definite conditions, the surface tension of a cast iron melt exerts a considerable influence on the graphitisation processes and hence on the form of the graphite.

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GERMANY

Travelling-Crane Runways and Rails in Steelworks

By K. ECKINGER, E. KRANZ, and H. P. WITT. (Abstract of three papers in *Stahl und Eisen*, Vol. 74, No. 22, October 21, 1954, pp. 1418-1437, 49 illustrations.)

OVERHEAD cranes in steelworks are expected to operate continuously with a minimum of maintenance. This can be achieved only by designing according to past experience of this type of crane, and by paying special attention to upkeep. The greatest wear is caused by the repeated load changes and the impact and lateral forces which arise.

Compared with solid plates, lattice-work girders are subject to much more rapid fatigue, since they suffer quadruple sign change of the deflection at a single traverse of the load. With riveted girders the wheel load has to be transmitted to the web through a number of rivets, which rapidly tend to become loose, and with welded girders any air gap between web plate and flange tends to overstress the rigid fillet weld. Consequently, care should be taken that the web plate comes under direct stress, by planing the edge of the web and by dressing the upper flange accurately. The depth of a good travelling-crane girder should be at least 10% of the span, to avoid too great a deflection. Perforated plates, which allow a wide gangway, have been found excellent as horizontal stiffeners.

Periodically, the runway of the crane, particularly in districts where soil is liable to subsidence, must be re-aligned. In this case it is inadvisable to grout the legs of open-air crane runways. The anchor supports alone are set in concrete and project well above the foot of the support, which can then easily be lifted by a hydraulic jack, tilted, and the base of the support filled in. If the runway is inside a shed, accessibility for re-alignment and maintenance should be taken into consideration when designing the layout. The shed and the crane runway should be designed as separate structures and should not be joined up by intermediate members.

Concerning various national rail standards, the crane rail standardised in Germany is the easiest to bolt, weld, or clamp to the girder, but the American rail, with narrow flange and deep web, is the easiest to roll. On the whole, the British type, with a thicker web than the American and a greater depth than the German rail, represents a good compromise between the requirements of the crane manufacturer and the rail manufacturer. For the lighter-duty steelworks crane, riveting or welding-on the rails has proved to be safe practice. In other cases it is advisable, where a break is present in the crane support, to provide an extended rail joint, which may be either of the stepped or mitred type. By shearing or planing off the flanges, the rail ends always remain fully supported. The best practice is for the crane rail to run continuously from one end to the other. For this purpose, however, the runway support of the crane must be developed as a continuous beam, but this is advisable only in areas which are free from soil subsidence. The crane rail should be kept in line by guides, which are riveted or welded to the upper flange of the girder. Friction between rail and bed plate can be reduced by means of graphite grease. Measurements of the stiffness of the girder flange plate have shown that the rails really act as part of the supporting girder, if they are continuously riveted in a correct manner; rail flanges must therefore be included as load-carrying members and their cross-section allowed for accordingly. Recently, types of fastenings have been used composed of laminated plastic between rails, supports, and bolts. Indentation of the rail ends into the support can be avoided by fitting interchangeable shims underneath the rail at these points.

In reinforced-concrete crane runways, the top reinforcement of the concrete tends to be located as near to the crane rail as possible. It is necessary to safeguard the concrete-bearing zone, and for this purpose, immediately before placing the concrete, flat steel plates at 50 cm pitch are welded onto the top compression reinforcement bars. These plates are bevelled at one side and shaped in such a way that the rail bolts with " swallow-tail " heads are inserted sideways into the slots so formed, and the rails are clamped tight. The elastic behaviour of the concrete compression area of the cross-section is better in respect of stress distribution than a monolithic concrete surface, which crumbles under high wheel loads. In the asbestos process, tie-bolts are used, which screw into preformed holes in asbestos bushes with a partly cut thread. Owing to the flexibility of the composite asbestos and concrete construction, the tie-bolts can deflect with the rail.

Most of the failures in crane rails on concrete runways originate from the interface between rail and concrete. In reinforced concrete a hair-line crack formation in the concrete is known to be distributed over the whole of the tensile zone, including that of the reinforcement tension bars. Although durability is not adversely affected by these hair-cracks, somewhat larger cracks have an influence on life, as they are points of attack for corrosion and promote flaking of the concrete. Crane rails should be clamped on or mounted to move longitudinally only. Concreting-in of double-T supports is a good safeguard. Special steel bearing plates must be placed underneath the rail butts and the concrete expansion slots bridged over by light steel plates. In such a structure there are no objections to clamped-on crane rails, which are thermit-welded along the whole run. As a whole, the efficiency of concrete crane runways depends primarily on building construction, particularly with regard to the joint between crane rail and concrete.

British Industrial Developments

Plastic Flexible Ducting for Ventilation

There can be few places where more is demanded of ventilating ducting than when it is installed underground in a mine. To fulfil its functions successfully and economically, it should be easily transportable underground and should be quick and easy to instal; it should possess and maintain a high efficiency; it should be non-inflammable, and should not support combustion; it should be rot-proof and unaffected by corrosion; it should be reasonably robust, but should be capable of quick and easy repair, should damage occur; and its cost must be low in relation to its life.

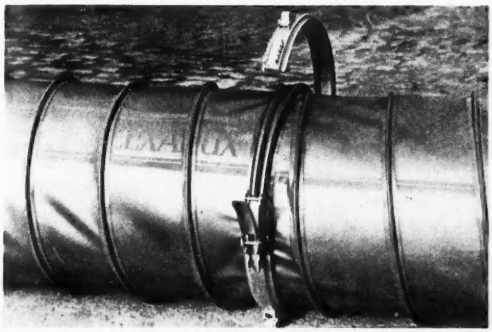


Fig. 1.

The two principal types of ducting in general use to-day are the metal and the fabric type of ducts, both of which possess certain well-known disadvantages. Thus, for instance, while the metal duct has a rigid and robust construction, it does not lend itself to easy transportation and installation. Deterioration occurs in the form of rust and of dents which increase friction losses; flanges are also strained, giving rise to quantity losses. These factors result in a substantial reduction in the initial high efficiency.

On the other hand, the fabric type overcomes some of these difficulties. It is very much easier to transport and much quicker and easier to instal. When, as is common, stiffening rings are placed in the bore, the efficiency of this type is, however, very much lowered because of the high frictional losses. This type of ducting is usually made of canvas fabric or woven jute, both of which are inflammable and liable to become brittle and to rot, although they can be afforded partial protection by proofing with natural or synthetic rubber.

In an endeavour to overcome these disadvantages, a plastic ducting, marketed under the trade name "Flexadux", has recently been developed on the Continent, where it is now used extensively. The manufacturing rights in the United Kingdom for this product are held by Wilkinson Rubber Linatex Limited, of Camberley, Surrey.

Flexadux is a highly flexible plastic ducting which can be collapsed into a very small space. This is extremely advantageous, both during transportation and also where storage space is limited. With the exception of steel coupling and stiffener rings, it consists of polyvinyl chloride (PVC) plasticised with tricresyl phosphate. This plastic material, which is 0·025 in. thick, is rot-proof and age-resistant. It will not support combustion and has a tensile strength of 3200 psi. It can be used in temperatures varying from — 25° to + 60° C. Ducts made from it are relatively immune to damage, owing to their flexibility. Should, however, damage be sustained, a repair can easily and quickly be effected by means of a simplified version of the same welding process used in the original manufacture of the ducts. To prevent corrosion, the steel coupling and stiffener rings are totally enclosed by a plastic covering, the stiffening rings being arranged externally on the duct, so as to provide a smooth uninterrupted flow passage through the bore.

To ensure an air-tight connection between the lengths of ducting, and thus minimise quantity losses, a patented hinge clip is used. The ends of the ducts, which are reinforced with steel coupling rings, are placed in the clip and pressure is applied by means of an adjustable lever integral with the clip, no tools being necessary. The clip can also be quickly released, enabling the ducts to be uncoupled for transfer to other working points (Fig. 1).

The following types of Flexadux are manufactured:—

Type "S":—This type, as shown in Fig. 2, incorporates a steel stiffening wire in continuous helical form, and is suitable for both positive and negative pressures up to 20 in. water gauge. Standard diameters are 8, 12, 16, 20, 24, 28, and 32 in. The standard length is 13 ft, although other lengths can be supplied. This type is suitable for both suction-type ventilation and for supplying air from a blower.

Fig. 2.

Type "B":—This type, in which the steel stiffening rings are circular, is suitable only for a positive pressure of 20 in. water gauge. Standard diameters are 8, 12, 16, 20, and 24 in. The standard length is 13 ft, but other lengths can be supplied. This type is suitable for use when blowing.

Flat Type:—This type is intended only for emergency use. No steel supporting rings are incorporated, and the ducts are coupled together by interlocking the end coupling rings. Standard diameters are 8 and 12 in., and lengths can be supplied up to 150 ft. It will withstand positive pressures up to 20 in. water gauge only.

Constant-Speed Drive for Aircraft Electrical-Supply Systems

The advantages of constant-frequency a.c. electrical systems over d.c. systems in aircraft have long been appreciated. Recent trends in design and more arduous operating conditions have accentuated these advantages. Of several possible methods of driving alternators, the most practicable system would appear to be the variable-ratio hydro-mechanical drive, in which a unit is interposed between each main propulsion engine and its alternator, accepting variable-speed shaft power from the engine and driving the alternator at constant speed.

In order to take full advantage of the progress already made in the development of this type of drive by the Sundstrand Company of U.S.A., the English Electric Company has concluded an agreement to manufacture units to Sundstrand design in the United Kingdom, and plans are already well under way to commence manufacture of a type of drive on which a considerable amount of experience has been built up in the U.S.A.

Basically, the Sundstrand drive is a positive-displacement hydraulic transmission, comprising a variable-stroke pump and a fixed-stroke motor combined in one cylinder assembly and arranged in such a way that only the power represented by the difference in input and output speeds is handled by the pump and motor.

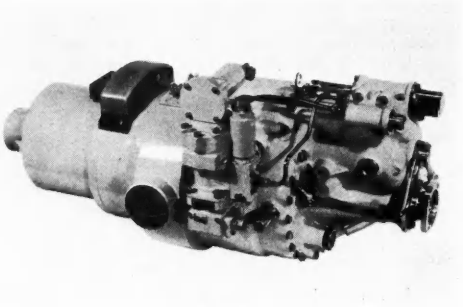


Fig. 1.

Several types have been developed and Fig. 1 shows a 40-kVA unit, of a type known as the package drive, directly coupled to an English Electric 40-kVA alternator. It is a similar type of drive to this that will be initially produced in Great Britain, and the following description applies to this type, which is designed to transmit 50 hp over an input-speed range of 3000 to 8300 rpm.

The " English Electric " Sundstrand constant-speed drive, shown schematically in Fig. 2, comprises a variable-displacement, axial-piston type of pump and a fixed-displacement, axial-piston type of motor, housed in a common cylinder-block assembly (6 and 10) and bolted together, with the port plate (8) positioned in between to direct oil flow between the pump and motor. Power from the aircraft engine is applied to the transmission by the input shaft and is transmitted through gears (13) to the rotating cylinder-block assembly.

Initial rotation of the input shaft simultaneously starts the cylinder-block assembly rotating, and the make-up and scavenging oil pumps (14 and 15). As the pump and motor are not self-priming, a gear-type make-up oil pump is used to deliver oil to the pump and motor assembly and the governor system. This make-up oil pump supplies the inlet of the main piston pump and motor so that the pistons (1) and push rods (2), will be held in contact with the output-controlling swash-plate assembly at all times, ensuring that a full charge of oil will be maintained in the high-pressure hydraulic circuit.

The variable-displacement pump consists of the cylinder-block assembly (6), the pump swash-plate assembly (4), and the manifold assembly (3). As the make-up pressure holds the pistons against the swash-plate assembly, piston stroking is accomplished when the cylinder block is rotated. The displacement of the pump is regulated by the angle of the swash plate, which is determined by the governor system through a hydraulic signal. The manifold assembly serves to deliver make-up oil to the pump and return oil to the pressurised case.

The fixed-displacement axial-piston motor consists of the motor cylinder-block assembly (10) and the swash-plate drive shaft assembly (11). The motor cylinder-block assembly receives high-pressure oil which forces the pistons to move out against the fixed-angle swash plate, thus converting axial-piston movement into rotary motion. The swash plate is free to rotate independently of the rotating cylinder block and hence can operate at speeds in excess of or below that of the rotating cylinder-block assembly. The offset of the motor valve plate is always in the same position with respect to the angular position of the swash plate, this being such that the motor push rods are exerting a clockwise torque upon the swash plate.

With the pump swash plate in maximum overdrive (maximum positive angle), the output shaft is overspeeding the input shaft. After gear-ratio corrections, this overspeed is the ratio of maximum pump displacement to the fixed displacement of the motor minus piston leakage. The swash plate remains in this maximum stroke position until output speed reaches 6000 rpm.

With a further increase in input speed, the normal tendency is to increase the output speed. At this point, however, the governor regulates the control piston (5) and swash-plate assembly to decrease pump-piston stroke. Less oil then flows from the pump to the motor and therefore the differential speed between the motor shaft and the cylinder-block assembly is reduced to retain constant output speed.

When the input speed exceeds output speed, the cylinder-block assembly will be turning faster than the motor shaft, which always operates at constant speed. Relative motion between the cylinder block and motor shaft will now be in a reverse direction, since it is now necessary to subtract from input speed to maintain governed output speed. Owing to the cylinder block overspeeding the motor shaft, the pump swash plate is held in an underdrive position (negative angle), permitting an oil flow from motor to pump. In effect, the pump is now motoring and the power flow in the hydraulic units is reversed.

Leakage from the cylinder-block assembly and lubricating oil which drains back into the unpressurised case are picked up by a gear-type scavenge oil pump (15) and returned to the external reservoir. The scavenge pump, which is attached to the same shaft as the charge-oil pump (14), is large enough to ensure a dry case condition at all times.

The alternator is driven through a full-complement, over-running, sprag-type, one-way clutch located in the output gear and clutch assembly (12) of the transmission. This clutch is necessary to prevent power from being directed through the transmission opposite to that from the main aircraft engine when the alternator is tending to act as a motor.

As the output speed has to be regulated to 6000 rpm constant, the speed signal for the governor system is taken from the output gear and clutch assembly. This signal is transmitted through the tachometer drive shaft and centrifugal switch to the governor assembly.

Initial rotation of the input shaft simultaneously starts the cylinder-block assembly rotating, as well as

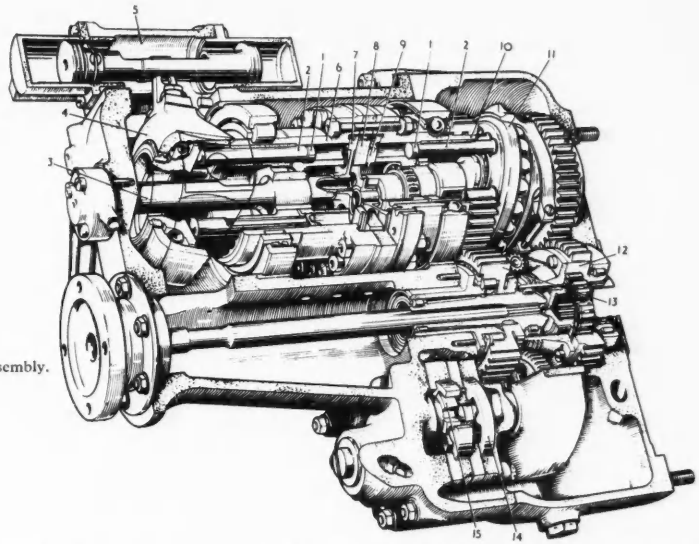


Fig. 2.

(1) Pistons.
(2) Push rods.
(3) Manifold assembly.
(4) Pump swash-plate assembly.
(5) Swash-plate control assembly.
(6) Pump cylinder block.
(7) Pump valve plate.
(8) Port plate.
(9) Motor valve plate.
(10) Motor cylinder block.
(11) Motor (fixed angle) swash plate.
(12) Output gear and free-wheel clutch assembly.
(13) Input gear.
(14) Charge pump.
(15) Scavenge pump.

the charge and scavenge oil pumps. The charge pump picks up oil from the external reservoir and delivers it through the oil filter and manifold assembly to the charge-oil section between the pump and motor cylinder blocks. Oil returns from the cylinder-block assembly through the manifold return line to the charge pressure-relief valve. This valve maintains the charge pressure at 300 psi, the excess dumping over the valve into the pressurised case. A by-pass taps off oil from the manifold return line and supplies the oil to the governor system and wobbler control pistons. The purpose of the pressurised case is to supply pressure lubrication to the various rotating parts of the transmission through properly located jet nozzles. Oil drains from the pressurised case to the unpressurised case through the lubricating-oil relief valve, which is adjusted to open at a pressure of 20 psi. Leakage from drains back into the unpressurised case is picked up by the scavenge pump and returned to the reservoir.

External to the drive, an oil reservoir, an oil filter, and an oil cooler are required. The cooler must be of adequate capacity to deal with the heat losses of the drive. A particularly attractive scheme in some installations is to integrate the Sundstrand oil system with the engine oil system, using a cooler of adequate capacity to deal with the heat rejection from both the engine and the constant-speed drive. The use of a fuel-cooled oil system often enables a very useful weight reduction to be effected.

The essential equipment associated with each alternator of a multi-alternator system may be itemised as follows:—

(1) The constant-speed drive, as described above.
(2) The control unit, to achieve " real " load sharing between parallel connected alternators and precise frequency control.
(3) The alternator.
(4) The voltage regulator and " reactive " load-sharing control.
(5) The alternator control panel, to provide adequate fault protection, etc.
(6) The main circuit breaker.
(7) The synchronising unit.
(8) The current transformers, for real and reactive load sharing and protective devices.

A block diagram, showing the arrangement of these components, is given in Fig. 3.

Additional equipment may be required, such as transformer-rectifier units to provide a 28-V d.c. supply for low-voltage services, such as lighting, control circuits, small actuators, etc.

Governor Control Units:—

The governor control circuits have two functions, i.e., to keep the alternator frequency constant and to equalise real load distribution between machines operating in parallel. If, owing to a change in engine speed or alternator loading, the frequency tends to drift from the correct value, a frequency discriminator circuit develops a rectified d.c. signal proportional to the deviation and of appropriate polarity, depending on whether the frequency is tending to increase or decrease. The signal is taken to a magnetic amplifier, which produces an equivalent a.c. signal, fed to one winding of the two-phase governor control motor, the other phase being supplied with a fixed phase voltage. The control signal leads or lags this voltage by 90 deg., and so causes the motor to rotate and adjust the governor control-spring setting to correct the frequency deviation.

If one of a number of alternators operating in parallel tends to lose speed, it will shed some of its real load, and a current transformer is used to utilise this effect to produce a signal in the drive control unit, causing the governor drive motor to adjust the drive output speed so that the alternator will take up its share of the load.

Alternator:—

The alternator is of the separately excited type, with its exciter built in as an integral part of the machine. The exciter is supported between the main bearings of the alternator, resulting in a compact complete assembly, and the characteristics of the alternator-exciter combination are such that a high short-circuit current is available for fault clearing. A further advantage of the separate exciter is the fact that it is immune from the effects of heavy overload or short-circuit and it is designed so that it is not demagnetised during these conditions.

Special consideration is given in the electrical design to limit harmonic content to a minimum and to provide a waveform which is satisfactory for all aircraft applications.

To enable the machine to operate at high temperature, glass insulation is used and the windings are

impregnated with silicone varnish. Iron losses are kept to a minimum by the use of high-grade low-loss core plate steel.

Voltage Regulator:—

The voltage regulator may be of the conventional carbon-pile type. If the voltage tends to move from the correct value, the action of the control coil adjusts the resistance of the exciter field circuit so that the deviation is corrected. The carbon-pile type of regulator has limitations which have led to the development of the magnetic-amplifier type of voltage regulator. The exciter field is connected to the output of the amplifier, the control winding of which receives a d.c. signal proportional to the alternator output.

If the voltage generated by one of a number of alternators tends to fall, the other machines will take up more of the reactive load of the system. A current transformer and a " mutual reactor " (a transformer requiring a large magnetising current, owing to an air-gap in the magnetic circuit) are used to cause the voltage regulator of the low-voltage machine to increase the excitation and so correct the deviation.

Alternator Control Panel:—

The functions of the control panel are to ensure that an alternator cannot be connected to the line if its frequency, voltage, or phase rotation is incorrect, or if there is a fault on the machine; it must also remove the machine if any of these conditions occur whilst the machine is running. The control panel incorporates devices which detect line-to-line or line-to-neutral faults up to the alternator circuit breaker; under- or over-voltage; phase sequence; and underspeed. Overspeed protection is arranged mechanically at the constant-speed drive.

Main Circuit Breaker:—

Each alternator is connected to its bus-bar by means of an " English Electric " type-681 circuit breaker. This unit is of the electrically closed and tripped type and employs double-break sintered contacts. It is rated at 120 A, with an interrupting capacity up to 4000 A at 60,000 ft altitude. The " closing " coil is automatically disconnected from its supply when the closing operation is complete, and the main contacts are held closed by a mechanical latch which can be released by the " opening " solenoid. Auxiliary contacts are provided for interlocking and indicating purposes.

Load Sharing:—

" Real " load sharing between parallel alternators and drives is accomplished by automatically adjusting the spring bias of the governor control associated with each drive. The spring bias is adjusted by means of a small geared two-phase induction motor which receives signals from a magnetic amplifier controlled by current transformers in each of the alternator circuits.

Associated Equipment:—

A complete range of a.c. utilisation equipment is being developed to enable full advantage to be taken of the reliability and weight saving offered by a.c. systems. These include motors, actuators, and transformers of various ratings and several of these items are already available. Motors range from 25 W to 5 hp output (or even larger, for particular duties).

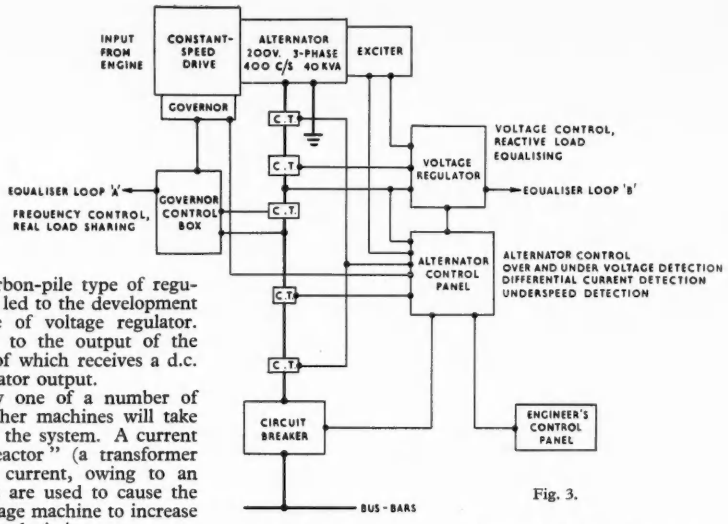


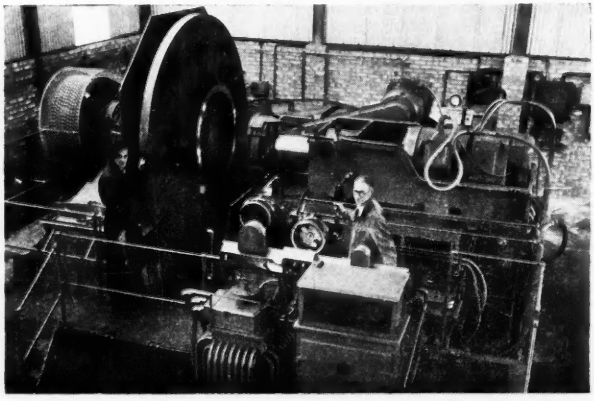
Fig. 3.

With regard to other types of hydro-mechanical drive, the cartridge type is fundamentally the same in operation as the package drive described above. This unit, however, is designed to be installed as an integral part of the main engine or gearbox and must therefore use engine oil and engine pumping, scavenging, and cooling facilities. This particular type of drive has not accumulated such a wealth of running time in service as the package type and English Electric are regarding this as the next stage in their constant-speed drive development programme. Investigation has shown that a lower total installed weight should be obtainable in a number of installations, with the cartridge-type drive.

108-in. Universal Grinding Machine

Stated to be one of the largest in the world, and the largest of its type on this side of the Atlantic, the 108-in. Universal Precision Grinder designed by Cooper Roller Bearings Co. Ltd., of King's Lynn, Norfolk, will accommodate and swing work of a maximum diameter of 10 ft 6 in. and a maximum length of 2 ft 7 in.

Grinding-wheel speeds are infinitely variable from 3500 to 6500 fpm by induction regulation of a.c. electric motors. Each wheel guard is fitted with a pre-set safety device, so arranged that the safe maximum speed



for any given wheel diameter cannot be exceeded. Wheel dressing, by pre-set diamonds, is automatic without appreciable loss of time. Two grinding-wheel spindles are provided, one for all facing operations and the other for all internal and external grinding, i.e., parallel, taper, and spherical.

Work speeds are also infinitely variable, by induction regulation of an a.c. electric motor, from 1·5 to 20 rpm. The work-head slide is capable of 5 ft 3 in. traverse by electric motor along its bed plate, to allow for different diameters of work. A scale calibrated in work and wheel diameters for both internal and external grinding enables the operator to position the work-head slide accurately, ready for grinding.

Coolant at the rate of 24 gpm is provided by each of two pumps, circulating through the spindle and through the grinding wheel to the work, thereby ensuring ample cooling, even during the heaviest grinding. Coolant from the machine is fed into an automatic filtering unit, which ensures perfectly clean coolant being pumped to the grinding wheels.

Hydraulic rams effect all slide movements and, in conjunction with a special "fine-feed" mechanism, enable accurate wheel advancement of 0·0001 in. to be made and also provide reciprocating movement to all slides. Regulating valves vary slide speeds from 1 to 48 ipm. When advancing the grinding wheel during grinding, the total weight of parts in motion is 18 tons, though these advancements can be made by hand without appreciable effort. Lubricating oil at 150 psi is automatically supplied to all slide-ways.

All electrical controls are grouped on duplicate console panels, so that the machine can be controlled comfortably from either side of the work. Each console panel is equipped with tachometers indicating work speeds and wheel speeds. The tachometer dials are calibrated to read rotational speed in rpm, peripheral speed of grinding wheel in fpm, and horse-power rating of motor, corresponding to the revolutions and the equivalent kilowatts input to the motor driving the grinding spindle in use. A kW-input meter provides a ready indication of the actual power input to the motor of the grinding spindle used. By comparing this with the reading of kilowatts available on the appropriate tachometer dial, the operator can determine how much of the motor power available is being used and thus keep the machine working to maximum capacity.

Adjustable mirrors are conveniently located so that the machine operator can observe the actual grinding contact. An automatic magnetic brake operates on the work-head when the "stop" button is used. Taper grinding is effected by swivelling the work-head. Approximate setting is by means of an electric motor, micro-adjustment being by hand. Maximum grinding wheel sizes are 24 × 6 in. for internal and external grinding, and 18 × 3 in. for face grinding.

Bright-Annealing Copper Wire for Electric Cable

A 75-kW bell furnace, manufactured by The General Electric Company Ltd., has recently been installed at the Eastleigh Works of Pirelli-General Cable Works Ltd. for bright-annealing copper wires in coils. The new furnace (Fig. 1) has three bases and hoods and the installation includes automatic temperature control and recording gear, and controlled-atmosphere plant.

The three bases, which are sunk into the ground, have heavy steel plate casings lined with heat-insulating brickwork. Coils of copper wire, with diameters up to 2 ft 6 in., are stacked to a height of 3 ft upon a raised plinth on a base and sealed within a cylindrical steel hood. The gas-tight seal between hood and base is effected by embedding the hood in a channel filled with a low-temperature melting alloy, initially melted with steam. After the hood has been purged of air, the

Fig. 1.

furnace bell is lowered over it (Fig. 2), the bell being accurately located in position by means of guide posts.

The heating time for a full charge, weighing with spools some 1800 lb, averages 1½ hr. During this time the metal seal is protected from overheating by a water flow. A fan mounted beneath the charge-carrying plinth circulates the protective atmosphere around the wire and ensures uniform heating. The charge is allowed to cool for a period of about 6 hr before being exposed to air.

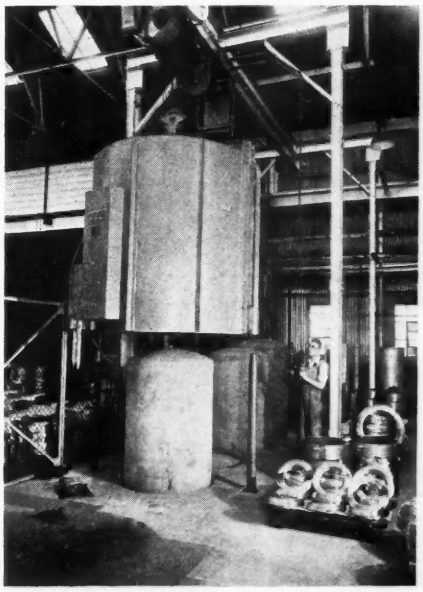
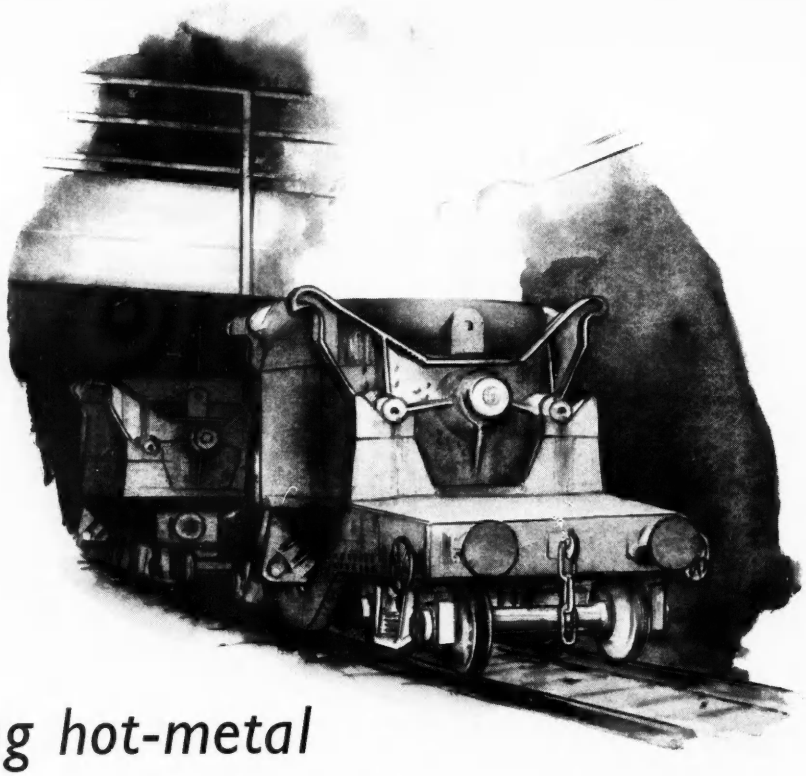


Fig. 2.

The furnace bell has an overall diameter of 6 ft 3 in., weighs 3½ tons, and provides temperatures up to 500° C. Its heating elements are of heavy-gauge nickel-chrome strip. The protective atmosphere is provided by a G.E.C. town-gas burner capable of delivering 300 cu ft of atmosphere per hour. The atmosphere is desulphurised before use

Protective devices are incorporated in the installation to prevent damage to a charge or to the elements by over-heating, and to prevent the removal of the furnace bell while the furnace is on load.



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CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

BEARING AND BEARING METALS

Some Experiments on Oscillating Bearings.

By F. T. BARWELL, A. A. MILNE, and J. S. WEBBER. (From a paper presented at a meeting of *The Institution of Engineers and Shipbuilders in Scotland, Glasgow*, January 25, 1955, 51 pages, 25 illustrations.)

As little information is available on this subject, the tests described in this paper were made on a 2-in. diameter bearing in a machine designed to provide variable load and frequency of operation. Tests on white-metal bearings led to the introduction of a grooving system; later tests were made on leaded bronze. Plain and compounded oils were used.

Among others, the following conclusions may be drawn from the results of this investigation:—

(1) As is the case for bearings generally, the operation of oscillating bearings may be divided into two phases, representing the running-in and "no-wear" conditions. Performance during the running-in period is shown to be markedly dependent on the nature of the bearing lining and lubricant. With certain combinations it was impossible to attain the no-wear condition under the severe test procedure employed in the investigation.

(2) Contrary to the case of bearings in which the shaft revolves continuously, the presence of oil grooves in the loaded region may improve lubrication.

(3) Form and disposition of oil grooves have a marked effect on performance.

(4) Under the conditions of load and speed applied during tests, bearings made from leaded bronze and lubricated with a compounded oil wore at a lower rate than white metal. Under favourable conditions, for example, after an initial wearing-in period, the rate of wear was negligible (less than 0·0001 in. in 500 hr).

(5) The maximum safe continuous intensity of loading, using a leaded-bronze bearing lubricated with compounded oil under force-fed conditions, was 1200 lb per sq in. of projected area.

(6) Under conditions where lubricant was fed by gravity, the intensity of loading at failure was 800 lb per sq in. of projected area.

(7) The use of leaded bronze or lead-base white metal appears to offer scope for improvement in top-end bearings of marine steam engines lubricated with compounded oil.

(8) There is no reason to believe that there would be any improvement in diesel-engine practice by the use of these materials in place of tin-base white metal.

(9) It is possible to obtain conditions at least analogous with hydrodynamic lubrication in bearings whose motion is intermittent.

HEAT EXCHANGERS

Theory of Moist-Air Heat Exchangers.

By G. BROWN. (From *Acta Polytechnica, Sweden*, Physics and Applied Mathematics Series, Vol. 2, No. 11, 1954, 33 pages, 5 illustrations.)

IN air conditioning, the problem of designing heat exchangers, in which the surface temperatures are wholly or partly lower than the dew point of the air, is often encountered. In such heat exchangers, the moisture contained in the air is partly condensed, and the water transfers a certain amount of heat over and above that quantity of heat which is transmitted when no condensation takes place. In heat exchangers of the most common type, air is cooled as it passes over coils or tubes carrying water, brine, or various refrigerants, but there are also other types of heat exchangers. For instance, the heat contained in warm, exhausted, moist air is sometimes utilised to heat cold, fresh air. Both in air coolers and in fresh-air heaters it is the air that is cooled, with the result that the moisture contained in the air undergoes condensation. In other cases, again, e.g., in evaporative condensers, the surface is warmer than the air, but is kept moist, so that heat is transferred by evaporated moisture from the warm surface to the cold air.

The purpose of this paper is to evolve a method of design of moist-air heat exchangers, so as to comply with the requirements that it shall be as correct as possible from a theoretical point of view, and that it shall be adaptable for practical use. Such a method is described for the calculation of these heat exchangers, on the basis of the values which hold good when heat exchange takes place in the absence of condensation and evaporation.

The temperature of the moist surface is determined by means of a graph. It is shown how the thermal efficiency of the fins is computed when the moist surface is provided with fins. In computing the size of the surface, a logarithmic difference in enthalpy is used. The dry-bulb temperature of the air at the outlet of the heat exchanger is determined by means of a graph in the Mollier $i - x$ chart. When it is not the entire surface of the air cooler that is moist, the condition of the air is determined at the boundary between the dry section and the moist section, and each section of the surface is computed separately.

INSULATING MATERIALS

Thermal Properties and Applications of High-Temperature Aircraft Insulation.

By P. GREEBLER. (From *Jet Propulsion, U.S.A.*, Vol. 24, No. 6, November - December, 1954, pp. 374-378, 3 illustrations.)

IN this article, experimental conductivity data on a fibrous insulation at various mean temperatures and densities are analysed, to determine the contribution by each of the mechanisms of heat transfer to the total conductivity. Radiation and air conduction account for nearly all the conductivity at higher temperatures. Conductivity due to radiation, which varies approximately as the cube of the absolute temperature, can be reduced by decreasing the pore size of the insulation, either by increasing the density or by reducing the fibre diameters. Increasing the thickness of a felt produces a proportional reduction in its thermal conductance, while an increase in the density of the felt produces less than a proportional reduction in conductance. An internal reflective foil does not appreciably lower the conductance of an insulating felt. Conductivity data on a high-temperature insulating felt are given as a function of mean temperature and density; data are used for heat-transfer calculations pertaining to jet-propelled aircraft.

High-temperature aircraft insulation generally con-

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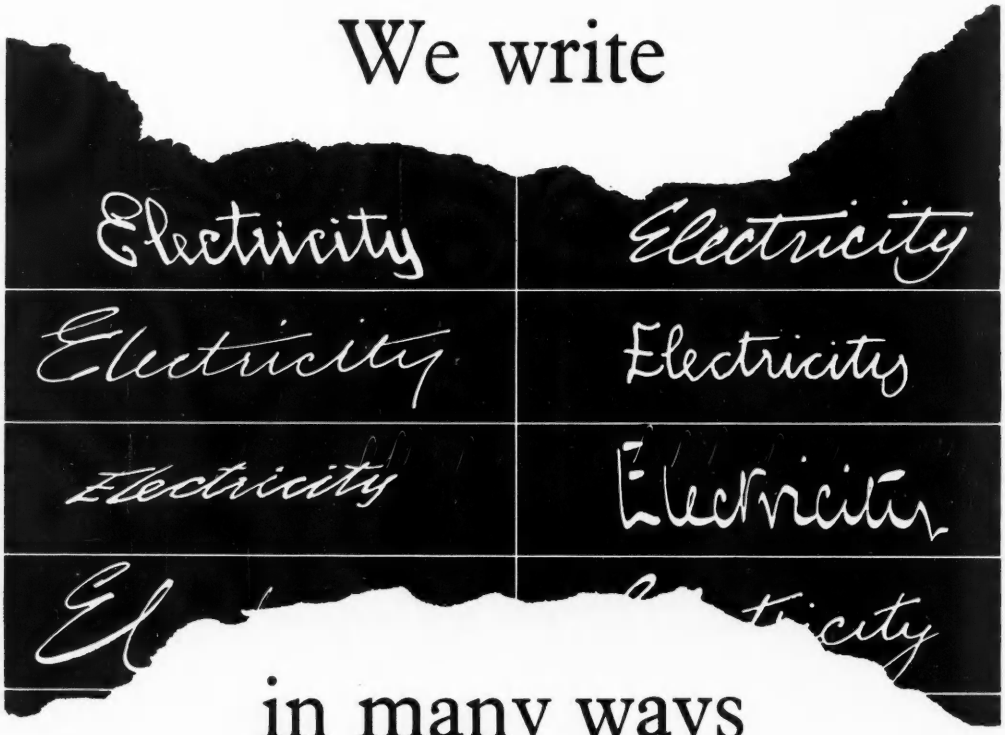

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sists of an alumino-silica or high-silica felt enclosed in metallic foils. The fibres of the felt, which is highly porous, do not soften or devitrify at temperatures in excess of 2000° F, and their diameters range from 1 to 10 microns. In nearly all applications, the air spaces occupy more than 90% of the total insulation volume. Heat transfer through the felt occurs by air conduction and convection, by radiation, and by fibre conduction. The purpose of the insulation is to break up a large air space into a great number of small pores, thereby reducing heat transfer by convection and radiation. If the insulation is used in series with an air gap, the emissivity of the foil may have an appreciable effect on the overall transmittance. In the application of a jet insulating blanket over a tailpipe, with still air between the cold foil of the blanket and the air frame, the air gap is estimated to contribute about 40% of the total thermal resistance from tailpipe to fuselage, with a foil emissivity of 0·3. For such applications of insulation, the performance of the insulation in a functional test should be evaluated from the temperature rise of the structural component being protected, or from the reduction of the cooling load required to limit the temperature rise to a specified value. The temperature attained by the cold foil of the insulating blanket is not a good indication of the performance of the insulation.

Thermal insulations are frequently used in jet-propelled aircraft to provide protection for structural components against excessive temperature rise under transient operating or emergency conditions. If there is sufficient space for a fairly thick insulation, a low thermal diffusivity of the insulation can provide adequate thermal protection, even though the heat capacity of the insulated component may be very low. If available space permits only a very thin insulation, the heat capacity of the insulation may generally be neglected, and the conductance is the only thermal property of the insulation that is effective in retarding heat transfer. Several problems are described, and solutions given with a minimum of mathematical detail, to illustrate the function of high-temperature aircraft insulations in steady-state and transient heat-flow applications.

INTERNAL COMBUSTION ENGINES

Choice of Cylinder-Bore Finish for the Running-in of Engines.

By K. R. WILLIAMS and S. G. DANIEL. (From a paper presented at a meeting of *The Institution of Mechanical Engineers, London*, February 8, 1955, 10 pages, 14 illustrations.)

WHEN a reciprocating internal combustion engine is first assembled, perfect congruence between the piston rings and cylinder bore does not exist. Minute deviations from a truly cylindrical form, brought about by imperfections in the machining of the piston rings and cylinder bores, coupled with distortion of these components due to various causes, prevent perfect congruence or mating of these parts in the newly built engine. When a newly assembled engine is first run, the piston rings will contact the cylinder bores only at certain points around their periphery.

During the process of running-in, the high spots on piston rings and cylinder bore are worn away, and running-in may be considered complete when mating of these surfaces has occurred. Since diametral wear of the piston ring usually occurs at a greater rate than that of the bore, it is normally the rate at which the bore wears that determines when the engine is run in. Although wear of the liner is relatively rapid at the ends of the piston-ring stroke, in the middle of the stroke wear occurs much more slowly. However, the engine is not completely run in until mating of rings and bore occurs throughout the stroke. Until the engine is fully run in, the load applied to the piston ring by its own tension and the cylinder pressure will be carried by a part of the ring periphery only. If the engine is run under high load conditions, before it has been fully run in those areas of the piston rings and cylinder bores which are actually carrying the load will be subjected to unduly high pressures. If these pressures are sufficient to disrupt the lubricant film between ring and bore, "scuffing" or "galling" of these components may occur.

This paper shows that an optimum surface finish exists which leads to rapid running-in when either straight mineral or additive-type engine oils are used. In particular, it is shown that the running-in of engines fitted with either cast-iron or chromium-plated piston rings may be of short duration when additive-type engine lubricants are used in conjunction with a suitable bore finish. Descriptions are given of rig and engine tests which demonstrate the influence of surface finish on both the wear and the consumption of oil occurring during the running-in process.

It is suggested that there are two components of surface finish which affect the running-in process. One of these components, the surface undulations of short wavelength (surface roughness), influences the rate of wear during break-in, whilst the surface undulations of longer wavelength (surface waviness) determine the amount of wear which must occur before the running-in process is complete. Whatever surface finish is chosen for the cylinder bore, to achieve as rapid running-in as possible, it is desirable to eliminate those errors in the machining process which give rise to waviness.

LUBRICATION

An Investigation of Heavy-Duty Lubricating Oils by Electron Microscopy.

By F. T. BARWELL, L. GRUNBERG, and D. SCOTT. (From an advance copy of a paper submitted to *The Institution of Mechanical Engineers, London*, January 1955, 7 pages, 15 illustrations.)

LUBRICATING oils containing heavy-duty (H.D.) additives are at present qualified on the results of prolonged and expensive engine tests, and every possibility of transferring part of this work to physical and chemical investigations appears to be worth exploring. In a recent symposium on the "Examination of Used Lubricating Oils" the results of exhaustive investigations were presented. The results reported indicated that, generally, the oil-insoluble "carbon," with which crankcase oils became contaminated in use, consisted of smoke particles formed during the combustion of the fuel in the cylinders. This was not applicable when combustion conditions were perfect, so that no smoke particles were formed, or when low viscosity index (V.I.) and naphthenic medium V.I. oils were used, giving "carbon" as products of their decomposition. The presence of H.D. additives was believed to delay flocculation by adsorption to the surface of the particles.

The visual examination of the oils investigated was restricted to the optical microscope and the size of the smallest "carbon" particles which could be observed was of the order of 0·4 micron (4000 A). The investigations showed that the particles had an average size of between 0·8 and 1·1 microns. It appeared possible, however, that details outside the range of the optical microscope had escaped detection and that a study of H.D. lubricating oils with the electron microscope might provide additional information, since the range of observation could be increased, enabling particles measuring little more than 100 Angström units to be detected.

Several unused and used heavy-duty (H.D.) oils were investigated by the electron microscope. Plain oils containing no additive and an oil containing an anti-

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wear additive were also investigated. No support was found for the suggestion that unused oils containing H.D. detergent additives are colloidal suspensions, as the micrographs were perfectly clear and no particles were found to be present. Two types of particle were found to be present in used H.D. and plain oils. One type of particle was of the order of 0·05 micron and tended to arrange itself in chain structures. The other type was very much larger, up to several microns in diameter. The two types of particle may have different origins.

It is concluded that the electron microscope is unlikely to provide a very useful means for the inspection of unused H.D. oils and the authors express doubts regarding the validity of an explanation of their successful use based solely on their deflocculating or dispersant action.

MARINE ENGINEERING

Some Effects of Blade-Section Shape on Model Ship-Screw Performance.

By A. SILVERLEAF and T. P. O'BRIEN. (From a paper presented at a meeting of the *North-East Coast Institution of Engineers and Shipbuilders, Newcastle-upon-Tyne*, January 28, 1955, pp. 123-162, 12 illustrations.)

THE main purpose of the work described in this paper was to study some effects of variations in blade-section shape on the performance of model screws by analysis and experiments on a group of screws, in which all the other design features remained unchanged. To some extent this might be described as an attempt to make comparative tests on propellers in a water tunnel, analogous with experiments made on aerofoils and wings in a wind tunnel.

Such comparative tests are not necessarily a good basis for decisions on ship-screw design problems, and any conclusions which may be drawn from them can only be applied to ship-screw design with reservations. Three points of doubt, among many others, are :— (1) These model tests were made in uniform flow conditions, while ship screws operate in irregular flows; (2) equivalent ship screws incorporating different blade-section shapes will often have different blade-area ratios; and (3) merchant-ship screws often operate in conditions in which cavitation occurs but does not affect the screw performance, and in such cases the extent and nature of cavitation damage, which depend strongly on the form of cavitation, can be significantly affected by blade-section shape, even when performance is unaffected.

With these reservations in mind, the following tentative conclusions may be stated:—

(a) The design method given by Hill,* and his form of curvature and tip-loss corrections, are reliable.

(b) The lift-camber relation quoted by Hill* for a parabolic mean line requires further investigation.

(c) Under non-cavitating conditions, provided that the blade sections used have good lift and drag characteristics, variations in section shape have little effect on efficiency. However, it is possible that some gains can be achieved by using sections near the root with their maximum thicknesses forward of mid-chord, and also by avoiding high cambers at these sections.

(d) It is doubtful whether adherence to the theoretical optimum radial thrust distribution is of importance.

(e) Under cavitating conditions, changes in blade-section shape can cause significant differences. In particular, constraints imposed by geometrical conditions, such as flat face sections or fixed camber ratio, are harmful.

(f) Comparison of thrust breakdown conditions, and thrust and efficiency reduction factors suggests that " shockless entry " designs are inferior to those with a small amount of positive incidence.

* J. G. Hill: " The Design of Propellers," *Soc. N.A. & Mar. E.*, Vol. 57, 1949.

NON-FERROUS METALS

Creep of High-Purity Nickel.

By W. D. JENKINS, T. G. DIGGES, and C. R. JOHNSON. (From *Journal of Research of the National Bureau of Standards, U.S.A.*, Vol. 53, No. 6, December 1954, pp. 329-352, 33 illustrations.)

NUMEROUS equations have been proposed by analysis of creep data as descriptive of the extension/time relationship in the creep of metallic materials. The equations are modified and new ones are proposed as experimental results are accumulated. To-date, however, no theoretical treatment of the complete extension/time relation is universally accepted, and there is still need for additional precise data from experiments carried out under accurately controlled conditions. The interrelation of strain hardening, recovery, and recrystallisation is not fully understood, and the important role of the thermal-mechanical history on creep behaviour has not been generally recognised. Any adequate physical theory of the mechanism of plastic flow in creep must account for the influence of all these factors.

The tests on high-purity nickel described in this paper were made as one phase of a project to study the mechanism of creep in high-purity polycrystalline metals and to evaluate the creep behaviour of these metals in comparison with binary alloys of such metals. The comprehensive programme includes a study of the creep characteristics of high-purity copper, high-purity nickel, and alloys of these two metals. The relatively low temperatures at which creep may occur in each of these metals, the unlimited solid solubility of the two metals in each other, the availability of commercial metal of high purity, and the wide industrial applications of both the metals and alloys made the binary system an attractive one for use in the present investigation.

This paper is concerned with the results of creep tests made on high-purity nickel at 300°, 700°, 900°, and 1,200° F, the influence of rate of loading to selected creep stresses on creep behaviour, and the effect of prior strain history on the tensile properties of the nickel at room temperature. Contour and hardness surveys were also made on specimens fractured in creep and on specimens fractured in tension at room temperature, and the microstructures were correlated with creep and tensile behaviour.

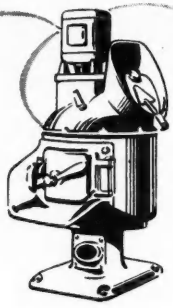
The experimental results are analysed in terms of the past and present theories of deformation of metals. Strain hardening, recovery, and the initiation, propagation, and types of fractures obtained during creep are further evaluated by means of true stress/true strain and hardness data obtained at room temperature and by metallographic examination of the fractured specimens.

PHOTOGRAPHIC TECHNIQUE

" Electrofax " Direct Electrophotographic Printing on Paper.

By C. J. YOUNG and H. G. GREIG. (From *RCA Review, U.S.A.*, Vol. 15, No. 4, December 1954, pp. 469-484, 9 illustrations.)

ADVANCES in materials showing photoelectric sensitivity are now pointing the way to new dry-processing photographic methods which are independent of silver halide chemistry. The methods depend on a photoconductive sheet which is electrically charged, exposed to produce a latent image, and then dusted with fine charged particles to make the image visible. The best-known



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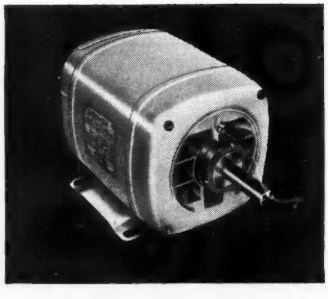

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system, called xerography, employs a selenium-coated metal plate, a pigmented powder for ink, and a transfer to paper for the final print. Commercially available phosphors and especially prepared cadmium sulphides have also been used to prepare plates for continuous-tone electrophotography.

The direct method described in this article is similar, except that the paper on which the final print is made is itself photosensitive. This " Electrofax " paper is ordinary paper with a thin flexible white coating of special zinc oxide in a resin binder. The basic " Electrofax " process can be described by the following simple procedure for making a direct print:—

(1) The paper is first made sensitive to light by giving it a blanket negative electrostatic charge on the coating side in the dark. One way of doing this is by ion transfer from a corona discharge.

(2) The sheet, now sensitive to light, is exposed by any of the conventional photographic procedures. The electrostatic charge is lost or reduced in the exposed areas and retained in the masked areas, to form a latent electrostatic charge image on the surface of the paper.

(3) The latent image is developed by applying a pigmented resin powder carrying a positive electrostatic charge. The powder is attracted and held by the negatively charged image areas.

(4) The powder image is fixed by melting the resin powder, so that it fuses to the paper surface to produce a durable light-fast image.

Even when these steps are carried out by hand, a finished copy can be produced in a fraction of a minute. With proper lighting, projection prints and camera exposures can be made in a similar way.

" Electrofax " paper is in the intrinsic silver halide photographic sensitivity range and is thousands of times more sensitive than blue print, diazotype, and bichromated colloid papers. To give a practical example, a contact print on white paper may be obtained with a 1- to 2-sec exposure, using as the light source two 4-watt white fluorescent lamps at 20 in. These papers are therefore well suited for projection printing and microfilm enlargement with continuous processing.

The article includes curves showing the electrical and spectral characteristics of the paper, with a brief description of how these measurements were made. Several methods of developing the electrostatic image are mentioned, with emphasis on the one which at present shows the most promise in mechanised printing.

SPRINGS

The Effect of Torsional Overstrain upon the Static Strength of Spring Steel.

By J. A. Pope and J. E. Andrew. (From an advance copy of a paper submitted to *The Institution of Mechanical Engineers, London*, January 1955, 18 pages, 23 illustrations.)

In the past, much investigatory work has been carried out on the subject of metal springs, but, so numerous and complex are the factors involved, that their influence on the resultant behaviour of springs has never been fully analysed.

By coiling a spring longer than its required final length, and " scragging " to closure, the surface fibres of the wire are stressed beyond their torsional elastic limit, resulting in residual stresses being induced at the surface after removal of the load. This is a means of increasing the load-carrying capacity of the final product. The scragging process, although apparently simple, is more complex than may be generally realised. The work reported in this paper is an attempt to explain the phenomena observed, and gives the results of an investigation into the effect of low-temperature heat treatment upon the static properties of a hardened and tempered spring steel which had been previously subjected to torsional overstrain.

The results show that the textural stresses may be relieved by low-temperature heat treatment without reducing the beneficial body stresses produced by the torsional overstrain. At large overstrains the textural stresses are so great that plastic deformation in the reverse direction takes place on unloading. Such deformations reduce the residual body stresses, and therefore the elastic limit strength, after low-temperature heat treatment. To prevent the plastic deformation on unloading, the specimens should be given the low-temperature heat-treatment before removal of the load. Tests on six springs showed that the results obtained from torsion specimens directly apply to springs.

The improvement in the elastic-limit strength after low-temperature heat treatment is considerable, about 33%, while the increase in 0·1% proof stress, although marked, is somewhat less and of the order of 8·5%.

After the low-temperature heat treatment the elastic limit strength range (i.e., the sum of the elastic-limit strengths in the positive and reverse directions) and the 0·1% proof-stress range are unaffected by the degree of overstrain. In other words, for the steel tested and within the range of strain examined, torsional overstrain does not work-harden the material, any increase in strength in one direction being offset by an equal reduction in strength in the reverse direction. The change of strengths in the two directions is approximately equal to the corresponding body stresses induced by the overstrain.

STRESS ANALYSIS

The Behaviour of Plain and Notched Steel Specimens under Static and Fatigue Loadings.

By J. W. Fitchie. (From an advance copy of a paper submitted to *The Institution of Mechanical Engineers, London*, January 1955, 15 pages, 18 illustrations.)

The main object of the work described in this paper was to determine the true stress field at the base of a notch in specimens when loaded within and beyond the elastic range. With this information, fatigue tests were then designed with the object of correlating results from plain specimens with those from notched specimens for fatigue cycles with and without a superimposed mean stress.

The elastic and plastic stresses at the roots of semi-circular circumferential notches in cylinders under static tensile loads have been determined experimentally from measured strains. The elastic stresses are in close agreement with those predicted mathematically by Neuber for hyperbolic notches. For notched specimens, the rate of increase of axial strain with load and the radius of curvature of the notch remain unchanged for both elastic and plastic loadings. Unloading from a plastic load causes the stresses to decrease at the same rate as they increased under the initial elastic loadings. The elastic unloading persists for a range equal to about twice the initial tensile elastic range.

Experiments on plain and notched specimens have also been performed under fatigue loadings. The results show that the limiting range of fatigue loadings is constant for loadings within the elastic range, but decreases in the plastic range with increasing maximum load of the cycle.

The Mises-Hencky function of the three principal stresses is shown to be suitable for determining quantitatively, for both plain and notched specimens not exhibiting size effects, the onset of plastic flow, the nature of the plastic stress-strain relations, the behaviour under fatigue stresses, and the variation of fatigue-stress behaviour with superimposed static tensile loads. More critical stress systems could be devised to differentiate between the various laws available for determining these quantities.

"I told him straight . . ."

said the Factory Manager

"*I told the Guv'nor straight*," he said, "*— if you want this lot out on time, I want more up-to-date handling equipment!*"

We murmured admiration for his fearlessness . . . then we told him we were from Yale & Towne; we'd been called in by "the Guv'nor" to see how we could help.

"*Well, you can see for yourself the trouble*," he said with a sweep of his hand. "*Hold-ups here — hold-ups there. We just can't keep the stuff moving.*"

We checked everything that was moved in, around and out of the factory. Our recommendations went to the Board.

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"*The way the Guv'nor's beaming*," said the Factory Manager, "*you'd think it was all his idea.*"

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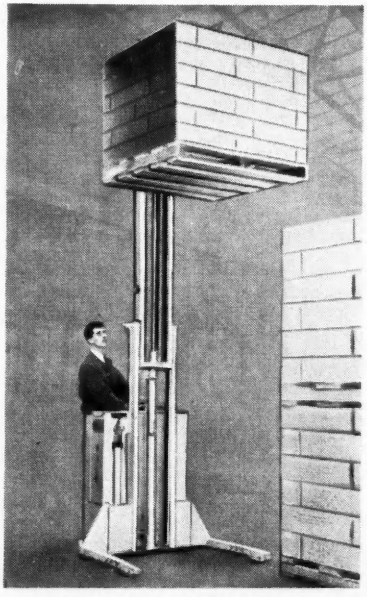

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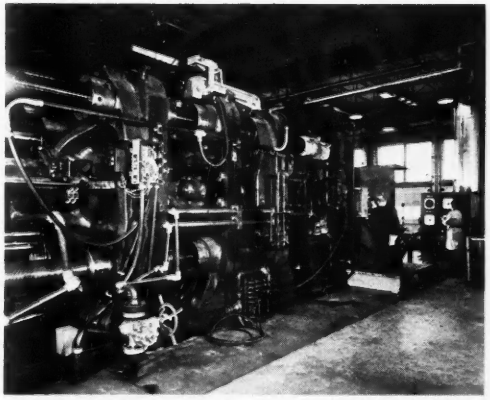
New Materials, Processes and Equipment

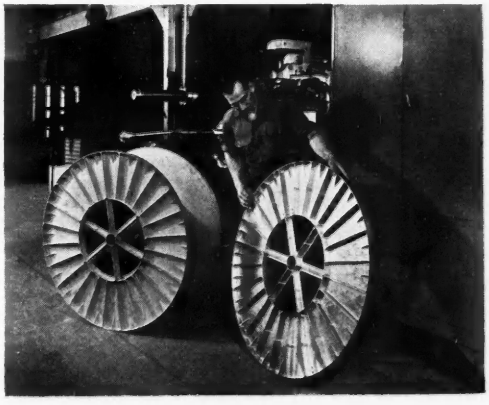
GIANT DIE-CASTING MACHINE

Aluminium castings weighing as much as 75 lb and zinc castings of 200 lb can be produced in a new die-casting machine, said to be the largest ever built, recently installed in a Toledo, Ohio, plant of the Doehler-Jarvis division of the National Lead Company. The machine, which was developed by Doehler-Jarvis and the Kaiser Aluminium & Chemical Corporation, weighs approximately 250 tons, and dies weighing up to 50 tons can be used in it.



The machine was designed primarily for aluminium casting, because the increasing weight of automobiles, due to larger bodies and the adoption of automatic transmissions, power brakes, and power steering, creates a difficult problem for the industry. In this connection, Doehler-Jarvis believe that the weight can be reduced materially by substituting die-cast aluminium engine blocks for cast-iron blocks and by using aluminium for door frames, instrument panels, rear-deck lids, and other large pieces now made of sheet steel.

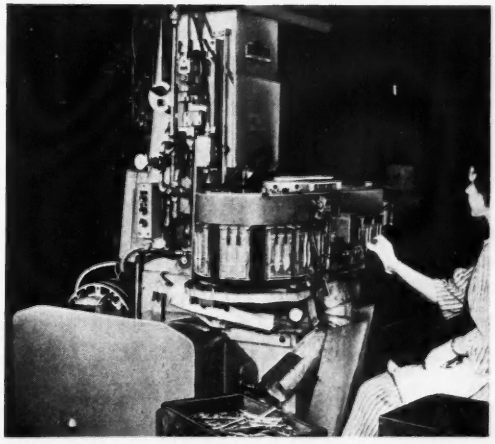
The machine can produce 75-lb aluminium castings at a rate of thirty per hour. As metal consumption at this high rate is estimated at 2250 lb per hour, an automatic ladle was developed to cope with this requirement. In addition, the machine is equipped with two 12,000-lb reverberatory furnaces. Successful casting temperatures are maintained by setting controls located on the metal ladle. Other special devices are an automatic or manual cycle control, automatic control of furnace temperature, and temperature recorders for the water-cooled sleeve of the casting cylinder and the dies. Some idea of its size is afforded by the illustrations, which show the machine and castings produced by it.



HIGH-PRODUCTION ABRASIVE-BELT GRINDER

A new abrasive-belt grinder, the " BG-8 Automat," for flat-grinding and polishing small metal parts rapidly and accurately, has been produced by the Engelberg Huller Company, of Syracuse, N.Y.

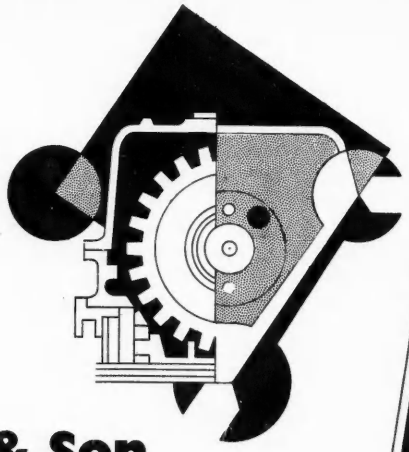
Workpieces for the new unit, which is equipped with a self-powered, continuous work feed and an 8×107-in. abrasive belt, are manually loaded but automatically ejected. Ejectors consist of either a positive stop to slide a part off its supporting ledge, or a spring claw entering a groove in a pad behind the work. For normal parts, the supporting ledge is shaped as a shallow V-block, with work supported close to the belt. The feed unit has 21 fixture plates, each $3\frac{3}{4} \times 6\frac{1}{4}$ in. The number of parts that can be loaded on each fixture plate is limited only by their size. Moved by parallel roller chains, each fixture plate is tracked to the leading edge of the abrasive belt, riding on hardened and ground shoes over hardened and ground steel rails. As the work advances to the belt surface, it is



fed slowly across the belt, and then retracted quickly before reaching the far edge of the belt. Feed speeds are variable; fixture-plate rates of travel can be set at 1090, 1350, or 1750 per hour. Depending upon the number of workpieces on each plate, production rates up to 14,000 or more pieces per hour are possible. The coolant system consists of a 40-gallon tank in the base of the unit, divided into two sediment zones, with a clean-out at one side. Inside the belt housing, a fan-shaped jet sprays the belt. Coolant is also delivered to a spray nozzle above the work area.

Two typical applications of the new machine are grinding 2100 knife handles per hour and finishing the sides of plier handles at the rate of 1440 parts per hour. Tolerances of $\pm$ 0·0005 in. to $\pm$ 0·001 in. were maintained for both jobs.

Smith's for Technical Books



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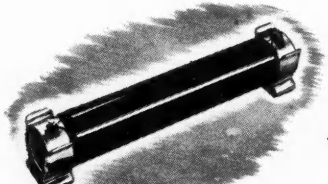
1 Perfectly even winding before and after firing.
Turns could easily become bunched during firing processes due to the surface tension of the molten enamel. Bunched windings would cause hotspots, burn-outs and breakdown of inter-turn insulation.

2 Moisture proof enamel and tube at all temperatures.
Cracks or pores in the tube or enamel would allow moisture to penetrate and cause corrosion, changes in resistance value and open circuits.

3 Enamel and tube must be good insulators at high temperatures of 300 to 400° C.
If the insulation resistance of the enamel or tube were to drop at high temperatures it would be necessary to impose a limit to the voltage which could be applied. This would reduce the permissible loading of units of high unit value, and increase the size or number of units required for a given duty.

4 Welded internal connections.
Imperfect internal connections would cause intermittent changes in resistance value and faults difficult to locate.

5 Enamel, tube and winding must be thoroughly annealed during firing.
Internal stresses in materials would be revealed as cracks in the surface of the enamel, cracked tubes or open circuits due to mechanical fracture of the fine wires.



6 Matched expansion coefficients of tube, wire and enamel.
The big operating temperature range of vitreous resistors would cause cracks in the ceramic materials or breakage of winding if stresses were set up due to unbalanced expansion coefficients.

7 The vitreous enamel must be chemically inert at temperatures up to 1,000° C.
Enamels which have a chemical action on the wire at high temperatures would cause eventual open circuits.

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5,000,000-Lb UNIVERSAL TESTING MACHINE

The Baldwin Locomotive Works, of West Eddystone, Pa., has now completed for the Aeronautical Structures Laboratory of the U.S. Naval Air Experimental Station at Philadelphia a 5,000,000-lb testing machine, stated to be the largest universal testing machine in the world.

The assembled machine stands 47 ft above laboratory floor level and extends 16 ft below. Around the outside of the machine is a cat-walk elevator with a capacity of 8000 lb, for use in preparing test set-ups and for observing results. Personnel performing tests are protected by a screen made of steel plate and bullet-resisting safety glass. Level with the laboratory floor, there will be constructed a transverse-bending test platform, 50 ft in length with girders 7 ft in depth, for testing beams horizontally under transverse bending loads. The total weight of the machine and the bending platform will be approximately 1,000,000 lb.

The machine will test specimens up to 30 ft long in tension or compression, and up to 50 ft long in bending. The clear space between the screws is 10 ft, and the length of the bed plate is 12 ft. The machine loads specimens hydraulically, the hydraulic loading cylinder being 54 in. in diameter and having a 36-in. stroke. Under a hydraulic pressure of 2500 psi, the machine will load specimens up to 5,000,000 lb; nevertheless, the weighing mechanism is so delicate that variations in load as low as 4 lb may be measured. The four large springs on the loading head completely eliminate the weight of any part of the machine from the indicated load on the specimen. The weighing system, which is entirely separate from the loading system, is guaranteed accurate to 0·5%. There are six loading ranges, 5,000,000, 2,000,000, 1,000,000, 240,000, and 24,000 lb. In testing long members in bending, it is necessary to support the specimen horizontally on a platform which is integral with the testing machine; otherwise, application of large downloads would lift the machine from its foundation.

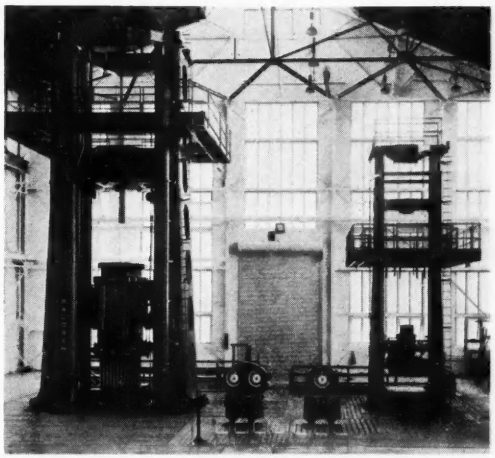
There are two platforms, one for the 5,000,000-lb machine and another for a 600,000-lb machine. Use of these platforms will permit the testing of specimens in bending up to lengths of 50 ft and 38 ft respectively. The platform for the smaller machine can withstand the full-load capacity of the machine at full span. For the larger machine, a load of 1,500,000 lb can be applied for the full 50-ft span, while proportionately higher loads can be applied as the span is decreased. It was deemed impractical to try to achieve full-load capacity for the larger platform as, even for the loading of 1,500,000 lb on the 50-ft span, two beams of 7 ft depth are required in the platform to obtain adequate rigidity. It is expected that this platform will provide sufficient capacity for the type of work contemplated.

The 5,000,000-lb machine consists essentially of three distinct systems, i.e., the loading, the weighing, and the control systems. In the loading system, the pump delivers oil under pressure to the hydraulic cylinder, which is forced down, taking with it the large screws and the sensitive crosshead. Since the tension crosshead at the top and the fixed bed at the bottom remain stationary, the downward movement of the sensitive crosshead applies either a compression or a tension load to a specimen. The weighing system, which is also hydraulic, is supplemented by a pneumatic system for extra sensitivity. The control system includes both mechanical and electrical units connected with the loading and weighing systems.

The most important unit of the weighing system is the Emery cell, which transmits load pressures to the indicator. It is essentially a shallow cylinder with a loose-fitting piston and a metal diaphragm so arranged that the load on the specimen produces a pressure change on a thin film of liquid trapped between the piston and the bottom of the cylinder. Relative movement of piston and cylinder is practically negligible.

Controls are all contained in a cabinet. Three large handwheels on the front of this cabinet provide control for application of the hydraulic load at any loading-head speed between 0 and 3 inches per minute, for instantaneous release of the load on a specimen, and for raising and lowering the loading head (for positioning purposes prior to testing), at speeds ranging from 0 to 20 ipm. The starter buttons above the handwheels control the various electric motors for raising or lowering the elevator, powering the large hydraulic pump, and turning the large screws for positioning the loading head. Knobs on the upper part of the cabinet control the rate of loading, the automatic maintaining of any predetermined constant load, and the selection of any one of the six load ranges from 24,000 to 5,000,000 lb.

Large dials on the cabinet indicate the load being exerted on a specimen, while smaller dials provide such information as air pressure, oil pressure in the backlash eliminators on the large screws, and the position of the loading head.



The machine will accommodate compression specimens of any height up to 360 in., neglecting strain in the specimen, and tension specimens of any length between 1 and 324 in. between grips, including strain in the specimen. The maximum tension length applies to visible tension space, to which may also be added the 36 in. of ram stroke, making a total of 360 in.

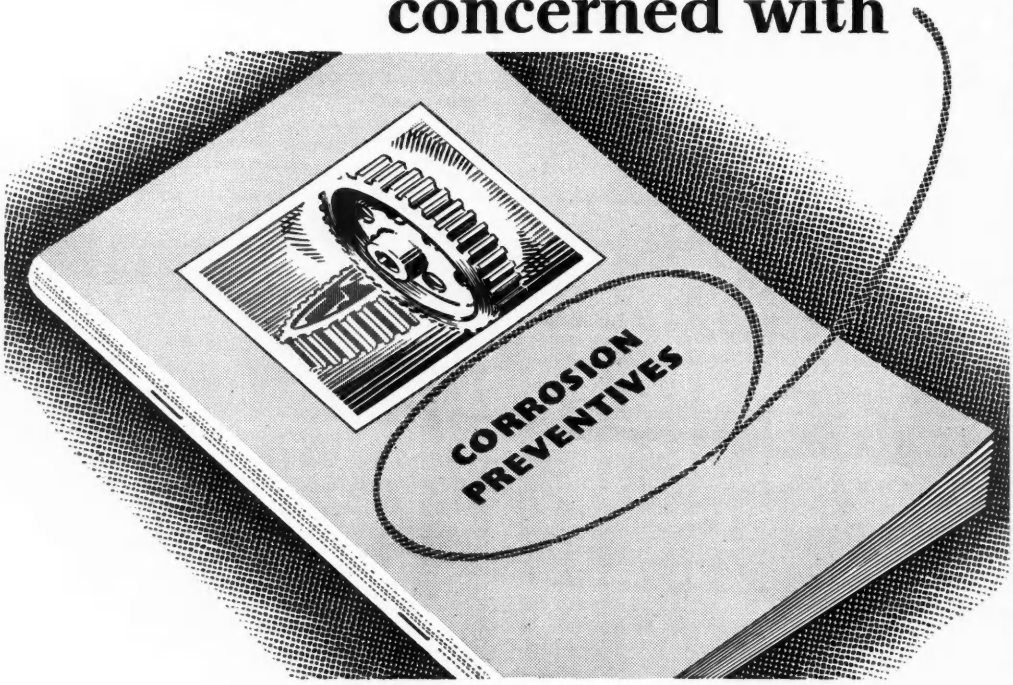
Some idea of its size is afforded by the illustration, which shows the new testing machine on the left, with a smaller machine on the right.

PORTABLE HYDRAULIC PLATFORM

In the welding of large drums of boiler shells, etc., considerable time is spent in erecting and dismantling the staging from which the operator works, often amounting to more time than it takes to carry out the welding assembly. In designing and producing new equipment to cut this time to a minimum and also to bring a greater degree of safety into this type of welding, Donald Ross & Partners Ltd., of London, N.W.1, claim to have produced in their " Twinner " Hydraulic Operators Platform the answer to this problem.

Staging is no longer required, as the new equipment is designed as a portable and robust platform, which can be positioned over the drum in a few seconds, permitting the operator to work with ease and safety. The platform, which has excellent forward reach, allowing the operator to work over the centre of drums 10 ft in diameter, is adjustable to any height from 5 ft 2 in. to 12 ft 4 in. The platform measures 4 ft 6 in. × 2 ft 3 in. and has a safe working load of 4 cwt in any position.

This Book is important to all concerned with



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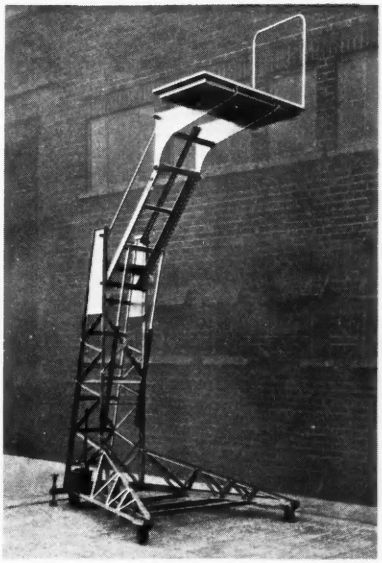
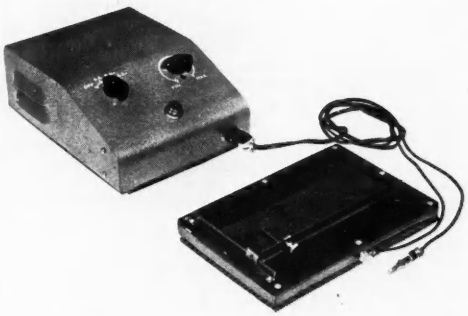


The latticed base is mounted on two fixed front wheels and two castors at the back, to facilitate easy steering into position. Two jacking screws at the rear of the base can be adjusted to ensure stable four-point suspension on the floor and to prevent coasting on unlevel surfaces. A hand-operated hydraulic pump immersed in a sealed tank of hydraulic fluid is also mounted on the rear of the base. Flexible hosing leads to two 30-in. stroke hydraulic jacks, pivoted at the rear bottom of the tower and also pivotally connected to the lower links of the platform. A relief valve is fitted on the pump. Steps on the tower are provided for climbing to the platform, which is floored with ¾-in. wooden boards. The total weight is 700 lb approximately.

In addition to its use in drum-welding operations, the new platform should be of interest to industry in general, where portable equipment is required for workmen operating considerably above floor level.

ELECTRO-CHEMICAL MARKER

A new electro-chemical marking tool for permanently etching into practically all metals anything which can be written, typed, or drawn, including intricate trademarks, has been developed by Crown Industrial Products Company, of South Bend, Indiana.

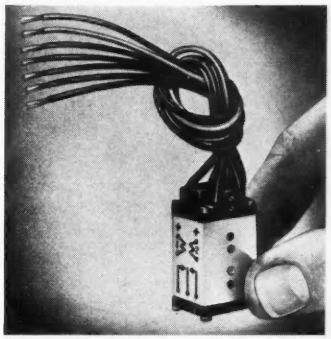


Designated " Crown E-Z Mark," the unit requires no special training or skill. Operation is simple and rapid, and no meters, high voltages, or acids are employed. The device, which may be called an electrolytic etcher, is completely portable, with separate power pack and marking attachments, and can be located on a bench or table, or even on the floor at a considerable distance from the stainless-steel marking tray, which holds the electrolytic fluid, sponge, and stencil. Adjustment controls are simple and, once set, need not be changed during a marking run. A guide bar ensures accurate location of the part on the stencil during production marking. The device can be used to mark steel, brass, copper, zinc, bronze, tin, and cadmium plating, and is even suitable for case-hardened and ground steel.

The part to be marked is placed on the prepared stencil, and momentary contact with the part is then made. The marking is etched permanently and clearly into the metal surface. Clarity of mark for various metals is controlled by a rheostat. Stencils can be used over and over again, for marking hundreds of parts.

MINIATURE MOVING-COIL RELAY

A miniature moving-coil relay, type 415/416, for applications in which sensitive control instrumentation is of importance, is announced by Electro Methods Ltd., of Stevenage, Herts. The relay consists of a balanced moving coil in the field of a permanent magnet. The coil has a contact leaf which operates between two adjustable contact springs mounted on the fixed frame. The coil assembly moves to one side or the other, and appropriate contact is made according to the direction of the controlling current. The coil has two independent windings which can be wired singly, in series, in parallel, or in differential, according to the application.



This relay fulfils the need for a magnetically self-shielded unit capable of operating from a power input of 10 microwatts and, as it is of balanced construction, functions efficiently in any position. The design is extremely compact, overall measurements being $1\frac{3}{16} \times \frac{3}{4} \times \frac{5}{8}$ in. The contacts are of platinum and may be employed to control a non-inductive load, provided that the a.c. or d.c. valves do not exceed 50 V, 100 mA, 2 W. Two models are available, type 415 with coils having a resistance of 100 ohms, and type 416 with coils of 350 ohms.

The sensitivity of these relays permits extremely close temperature control when used with a resistance thermometer in a bridge circuit. Similarly, such applications as self-balancing bridges will give close control without the use of an additional electronic amplifier. Other applications include transmitter-receiver muting circuits, instrument overloads, and direct on-off control from a light-sensitive barrier-layer cell.

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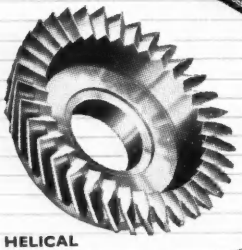
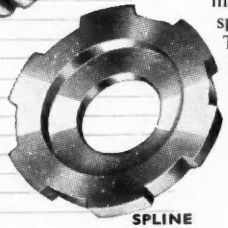


HELICAL DISC TYPE



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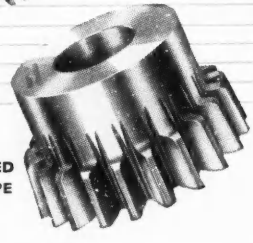

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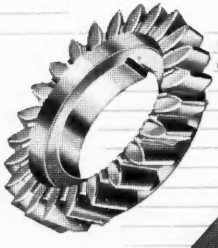

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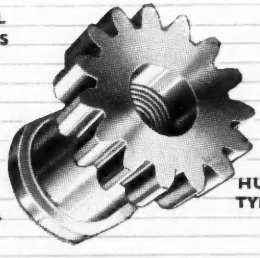

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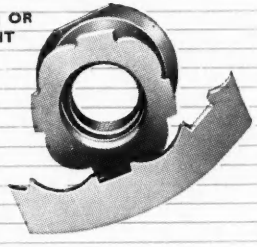

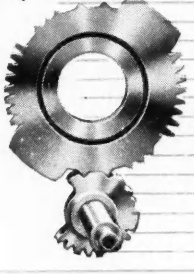

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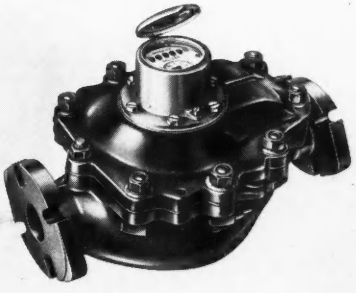
WATER METERS

A new range of water meters for trade supplies has been developed by George Kent Ltd., of Luton, Beds. These new meters, designated the " Master " range, are available in 1½, 2, 3, and 4-in. sizes. and are of the pipe-line, semi-positive type, intended to meet the need for accurate measurement of water flow at temperatures not exceeding 120° F.

Outstanding features of the range include:—Greater discharge capacity for a given size; large reduction in weight; improved straight-reading counter, reading in any desired units; lower head loss for a given flow; separate and replaceable working chamber and gear train; overall accuracy within 2%; improved frictionless gland, withstanding higher pressures; body lengths and flanges interchangeable with previous " MT "-type meters in all sizes; and extensive use of durable, corrosion-resisting materials, ensuring long working life and low maintenance cost.

The meter is built up of five main components, i.e., the outer casing or body, the measuring chamber, the under- or reducing gear, the counter, and the counter box. The body and cover are high-grade grey cast iron, treated with bitumastic solution to prevent corrosion. The cover, when bolted on to the body, clamps the measuring chamber in its correct position. Arrows on the cover and on the side of the body indicate the correct direction of flow through the meter. The measuring chamber and its top plate are machined from cast bronze, while the piston, and thimble inside, are of ebonite. A nickel-plated copper strainer in the inlet pipe prevents foreign matter from entering the meter. The undergear is common to all sizes, and the gears are made of nickel, as the train is submerged in water when working. The piston spindle and hub spindle are also of nickel. The counter is of cyclometer pattern, the counter-gear number rollers being made in non-corrodible white plastic, with figures in black. A pointer is provided for the indication of small quantities. Change-wheel facilities are accommodated in the counter unit. The dial of the meter is vitreous-enamelled copper, with indestructible markings and excellent legibility. The counter box is a brass stamping.



The 1½, 2, and 3 in. sizes have a maximum counter registration of 10 million Imperial gallons, while the 4-in. has a maximum of 100 million Imperial gallons. Meters can be calibrated in Imperial gallons, litres, cubic feet, or any other desired units.

A TECHNIQUE FOR CORROSION TESTING IN LIQUID LEAD

(*Concluded from page* 65)

divided into two compartments by fritted glass filter discs, and each compartment was connected by hydrogen and vacuum lines through a side arm. With the tube filled with hydrogen, the lead in each section of the loop was melted and, by the proper manipulation of hydrogen pressure and vacuum, the molten lead was then filtered into the bottom compartment. In this manner any mass-transferred material could be recovered and its position in the loop located.

The mass-transferred material in every loop which was operated was found to have collected in the bend at the bottom of the cold leg. In no case was there evidence of appreciable deposition on the cold-leg specimen. These facts were taken to indicate that most of the material was deposited in the cold leg through nucleation in the bulk of the liquid lead rather than at the walls. Plug formation appeared to be a matter of mechanical agglomeration of material in the bend at the bottom of the cold leg.

Although a large number of metals and alloys has been tested by this method, only the results obtained with Type 304 stainless steel are presented here.

Fig. 3. **Photomicrograph of plug taken from cold leg of lead. (× 3)**
Type 304 stainless steel, loop No. 2.

Two loops containing Type 304 stainless-steel test specimens were operated. The hot- and cold-leg temperatures were 805 and 490° C respectively for loop No. 1, and 800 and 450° C in the case of loop No. 2. Sufficient mass-transferred material to stop circulation of the lead collected in the cold leg of the first loop in 111 hours. The second loop plugged after 86 hours of operation. Fig. 3 is a photomicrograph of the plug taken from the cold leg of Loop No. 2. As indicated previously, this plug consisted of a mass of fine needles and platelets wedged together at the bottom of the cold leg of the loop.

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(1) H. N. Hackett: *Trans. Am. Soc. Mech. Engrs.*, **64**, 647 (1942).
(2) R. N. Lyon (ed.): Liquid Metals Handbook, pp. 1-28, Government Printing Office, Washington, D.C., 1950.
(3) R. C. Reid: *Mech. Eng.*, **73**, 514 (1951).

Mazda

High-Bay Fitting

This Mazda high-bay tungsten lighting fitting can be used either alone or mounted on a control box with one or two mercury vapour components to form a Mazda Blended Light Unit. Convected airflow minimises dust accumulation inside the reflector and does away with the need for a cover-glass. Only occasional cleaning is necessary.

SPECIFICATION

Spun aluminium *overlamp* concentrating reflector, electrolytically brightened and anodised. Simple pre-focusing adjustment for appropriate lamps, with panel indicator. Convected airflow vents. Cool wiring. Funnel finished in vitreous grey enamel. Conduit suspension $\frac{3}{4}$" tapped. Approximate weight (complete) : 7 lb.

Mazda

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B T-H

The Mazda C84816 Industrial High-Bay Fitting with anodised aluminium Reflector for use with one 500 W, 750 W, 1000 W or 1500 W G.L.S. Lamp.

Exploded view of funnel showing focusing adjustment and the method used to ensure cool wiring.

Write to your nearest BTH office for Descriptive Leaflet 9719-1 containing full details.

M4445C

NEWS OF THE MONTH

PERSONAL

Mr. T. H. Arnold, F.I.M., has been elected President of the Sheffield Metallurgical Association, in succession to **Mr. F. H. Saniter, B.Eng.**

Mr. C. J. Atkins has been appointed sales manager, **Mr. A. H. Woodley,** assistant sales manager, **Mr. F. W. Goodge,** contracts manager, and **Mr. D. Humphrey,** assistant to the chief engineer, of Keith Blackman Ltd., Mill Mead Road, London, N.17.

Mr. A. J. Beanland, general manager of the Trafford Parks Works, and **Mr. C. F. Jackson,** general manager of the Willesden Works of Lancashire Dynamo & Crypto Ltd., have been elected to the Board of the company.

Captain (E) Sir John R. Coote, Bt., C.B.E., D.S.C., R.N., has been promoted to be Rear-Admiral (E), and will relieve **Rear-Admiral (E) Sir Alex. D. McGlashan, K.B.E., C.B., D.S.O.,** as Deputy Director of Naval Ordnance (Material) and chief ordnance engineer officer.

Mr. G. H. Dowty, F.R.Ae.S., chairman and joint managing director of Dowty Equipment Ltd., Cheltenham, has been elected a honorary freeman of the Borough of Cheltenham.

Mr. H. de C. Falle, O.B.E., A.M.I.E.E., has relinquished his position of general sales manager of Aberdare Cables Ltd., 19 Woburn Place, London, W.C.1, after 50 years' service in the electrical industry. He will remain on the Board of Aberdare Cables (Holdings) Ltd., Aberdare Cables Ltd. and South Wales Switchgear Ltd. **Mr. W. R. Hall,** export manager, will continue to operate from the London office. **Mr. C. A. Hutchinson, M.I.E.E.,** will be responsible for home sales.

Mr. C. F. J. Francis-Carter, O.B.E., general manager of the Aluminium Wire and Cable Co. Ltd., Port Tennant, Swansea, since its inception in 1947, has been appointed managing director of the company.

Mr. Darby Haddon has been elected chairman of W. H. Dorman & Co. Ltd., Stafford, in succession to **Mr. H. F. Smallwood,** who is retiring from his position as chairman and managing director, but will continue to serve the company as technical consultant. **Mr. J. W. Whinpenny,** a director of the company, has been appointed to the position of general manager. **Mr. G. S. Tucker** has been appointed secretary, in succession to the retiring **Mr. G. C. Saunders.**

Mr. G. E. H. Hall, A.M.I.Prod.E., general manager, has been elected to the Board of Sheepbridge Stokes Ltd. and British Van der Horst Ltd., Chesterfield. **Mr. R. Gore, A.M.I.Mech.E., A.M.I.Prod.E.,** is now general manager of the Engineering Division, and **Mr. C. Hand** has been appointed general manager of the Foundry Division of Sheepbridge Equipment Ltd. **Mr. J. R. Jenkins** has been appointed methods engineer for the Southern Group of the Sheepbridge Companies, comprising Automotive Engineering Ltd., Hardinge Machine Tools Ltd., Light Productions Ltd., Clews Petersen Ltd. and Sheepbridge Engine Repairs Ltd.

Mr. G. W. Hall has been appointed assistant managing director, in addition to **Mr. L. Massey Hilton,** of The Fairey Aviation Co. Ltd., 24 Bruton Street, London, W.1.

Mr. Charles Hemm has been appointed director of projects (general rubber goods) of Dunlop Ltd. Mr. Hemm has been general works manager and a local director of Dunlop's factory at Cambridge Street, Manchester.

Mr. D. C. Kiernan has been appointed sales manager of Ormerod Engineers Ltd., Shawclough, Rochdale.

Mr. A. E. Lewis, formerly senior planning engineer at the Leavesden factory of the de Havilland Aircraft Co. Ltd., has been appointed works manager and assistant to the managing director of Daniel Smith Ltd., Castle Iron Works, Wolverhampton.

Mr. Thomas McIver has been appointed general manager of the Wallsend shipyard of Swan, Hunter and Wigham Richardson Ltd.

Mr. A. L. Markham has been appointed works manager of the new factory of Commer Cars Ltd., at Dunstable.

Mr. N. F. Patterson, director and general manager of Monsanto Chemicals Ltd., 8 Waterloo Place, London, S.W.1, has been appointed director responsible for all the company's manufacturing operations.

Mr. F. Pickworth has been elected chairman, and **Mr. F. S. Beale,** a director of the five newly formed subsidiary companies of English Steel Corporation Ltd., Sheffield, the English Steel Forge and Engineering Corporation Ltd., English Steel Rolling Mills Corporation Ltd., English Steel Castings Corporation Ltd., English Steel Spring Corporation Ltd. and English Steel Tool Corporation Ltd. **Mr. W. D. Pugh** has been elected managing director of English Steel Forge and Engineering Corporation Ltd. and a director of English Steel Rolling Mills Corporation Ltd., English Steel Castings Corporation Ltd. and English Steel Spring Corporation Ltd. **Mr. R. G. H. Taylor** has been elected managing director of English Steel Rolling Mill Corporation Ltd. and deputy chairman of English Steel Tool Corporation Ltd. **Dr. C. J. Dadswell** has been elected managing director of English Steel Castings Corporation Ltd. and English Steel Spring Corporation Ltd., and director of English Steel Forge and Engineering Corporation Ltd. **Mr. H. Scholey** has been elected director and general manager of English Steel Rolling Mill Corporation Ltd. **Major-General E. P. Readman** has been elected managing director of English Steel Tool Corporation Ltd. **Mr. W. C. Muirhead** has been elected to the Board of English Steel Castings Corporation Ltd. and English Steel Spring Corporation Ltd. **Dr. H. H. Burton, Mr. G. Gilfillan** and **Mr. W. E. A. Redfearn** have been elected to the Board of English Steel Forge and Engineering Corporation Ltd.

Mr. S. J. Preston, M.A.(Cantab.), A.M.I.E.E., chief executive officer of the Patent Department of Electrical and Musical Industries Ltd., has been elected chairman of the Radio Communication and Electronic Engineering Association, in succession to **Mr. C. G. White.**

Mr. C. G. Pullin, M.I.Mech.E., has been appointed technical consultant and director of Winget Ltd., Rochester. **Mr. Norman Staff** has been appointed director and secretary of the company.

Mr. Wallace Redpath, lecturer in the Department of Mechanical Engineering, Paisley Technical College, has taken up appointment as senior lecturer in the Faculty of Mechanical Engineering at the University, Rangoon, Burma.

Dr. G. W. Rigby has been appointed European technical representative of E.I. du Pont de Nemours & Co., Inc., Wilmington, Delaware, U.S.A.

Mr. H. L. Satchell, M.B.E., a director of The British Thomson-Houston Co. Ltd., Rugby, has been appointed deputy director of manufacture. On 1st March, 1955, Mr. Satchell will relinquish the managership of the Rugby works in favour of **Mr. H. E. Cox, M.I.E.E.,** who has been manager of the BTH Willesden works since 1946.

Mr. John V. Sheffield has been elected chairman, and **Mr. O. C. Kerrison,** managing director, of the Paterson Engineering

$$\frac{d^2\theta}{dx^2}+\frac{1}{x}\frac{d\theta}{dx}-\frac{\theta}{x^2}\epsilon=-\frac{V}{D}$$

$$D\left(\frac{d\theta}{dx}+\frac{1}{m}\cdot\frac{\theta}{x}\right)=M_R$$

$$D\left(\frac{\theta}{2}+\frac{1}{m}\frac{d\theta}{dx}\right)=M_H$$

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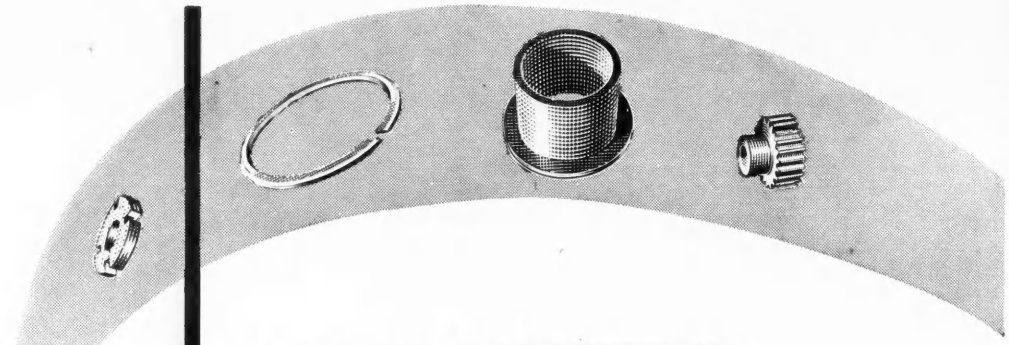

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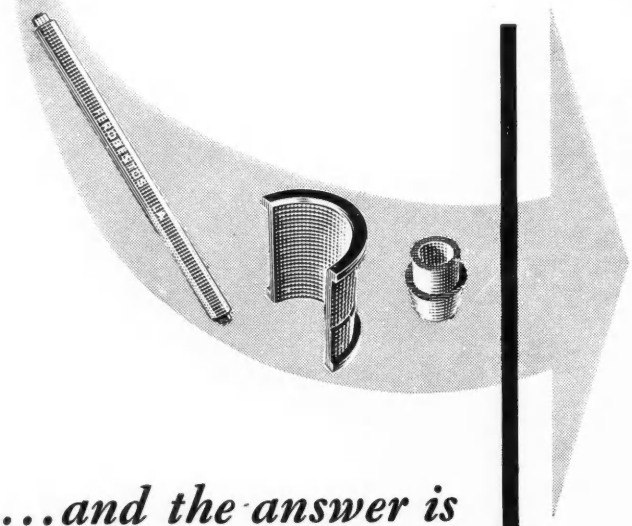

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FERODO LIMITED CHAPEL-EN-LE-FRITH · *A Member of the Turner & Newall Organisation*

Co. Ltd., Windsor House, Kingsway, London, W.C.2. Mr. S. W. Burrells has been appointed secretary of the company. **Sir William Paterson,** chairman and managing director, who founded the company fifty-three years' ago, has now retired.

Mr. W. Lewis Smith has been elected chairman of the Cable Makers' Association, **Mr. R. A. Bebb,** chairman of the Mains Cable Manufacturers' Association, **Mr. H. E. Helwig,** chairman of the Mains Cable Manufacturers' Association (Super Tension), for the second year, **Mr. H. Lockett,** chairman of the Overseas Rubber Cable Manufacturers' Association, and **Mr. H. H. Townsend,** chairman of the Rubber and Thermoplastic Cable Manufacturers' Association.

Mr. George Talbot has been appointed general manager of the Furnace Building and Contracting Division of The Wellman Smith Owen Engineering Corporation Ltd., Parnell House, Wilton Road, London, S.W.1.

Mr. B. J. Terrell has been appointed senior designer at the Pametrada Research Station at Wallsend. **Mr. R. F. Darling** has succeeded Mr. Terrell as senior research engineer, and **Mr. M. H. Petty** has been appointed deputy senior research engineer.

Mr. Joseph Walton has retired from his position of joint managing director of Thos. W. Ward Ltd., Albion Works, Sheffield, but remains available in a consultative capacity and retains his position as chairman and managing director of the Darlington Railway Plant and Foundry Co. Ltd., Darlington.

Mr. C. Watts, B.A., B.Sc., has been appointed foundry sales manager of the Distington Engineering Co. Ltd., Workington.

Mr. C. J. Williams has been appointed to succeed **Mr. J. A. Wellings** as manager of the Casting Division of High Duty Alloys Ltd., Slough.

ELECTRICAL POWER CONVENTION 1955

" Electricity and the Future " will be the theme of the seventh British Electrical Power Convention which is to be held at Brighton from 27th June to 1st July, 1955. Delegates from 38 electrical organisations from all parts of Great Britain, Northern Ireland, Isle of Man, and the Channel Islands will meet in the Dome, under the presidency of Sir Harry Railing, D.Eng., Hon.M.I.E.E., chairman and joint managing director of the General Electric Co. Ltd.

Associated with the Convention, a large and most comprehensive electrical exhibition will be held in the Sports Stadium, adjoining to which a display of electrical catering equipment will be staged; there will also be cooking demonstrations organised by the South Eastern Electricity Board.

The Exhibition will be officially opened by the President, Sir Harry Railing, on the night of Monday, 27th June.

The Convention will be opened on Tuesday, 28th June, with a civic welcome by the Mayor of Brighton, followed by Sir Harry Railing's presidential address. This will conclude Convention proceedings for the day—leaving the delegates free to visit the Exhibition. The annual luncheon of the Electrical Association for Women will be held on Tuesday; in the evening the delegates and their ladies will be the guests of the Mayor and Mayoress of Brighton at a reception.

A ten-year forecast on " The Generation of Electricity in Britain " will be given on Wednesday 29th June, by Mr. J. Eccles, C.B.E., B.Sc., M.I.E.E., M.Inst.C.E., M.I.Mech.E., deputy chairman (Operations) Central Electricity Authority (as the British Electricity Authority will be called after 1st April." " The Electrical Industry as a Career " is the subject of three papers on Wednesday afternoon. Sir Henry Self, K.C.B., K.C.M.G., K.B.E., Comp.I.E.E., deputy chairman (Administration), Central Electricity Authority, and chairman of the Education and Training Committee of the Electricity Supply Industry will deal with that industry's aspect of the subject; Mr. S. E. Goodall, M.Sc.(Eng.), M.I.E.E., vice-president of the Institution of Electrical Engineers and chief engineer of W. T. Henley's Telegraph Works Co. Ltd., will deal with " The Manufacturing Industry," and Mr. R. A. Marryat, B.Sc., A.M.I.E.E., vice-president of The Electrical Contractors Association (Incorp.), will speak on " The Installation Contracting Industry."

" Electricity and the Industrial Future " is the subject for discussion on Thursday, 30th June. There will be four papers, the first of which " The Use of Electricity in the Steel Industry " will be by a spokesman of that industry. Mr. B. L. Metcalf, M.I.E.E., M.I.Mech.E., chief electrical and mechanical engineer of the National Coal Board will speak on " The Use of Electricity in the Coal Industry." In the afternoon " The Use of Electricity in the Chemical Industry " will be dealt with by Mr. G. Nonhebel, B.A., B.Sc., F.R.I.C., F.Inst.F., fuel technologist and Head of the Fuel Economy Section of Imperial Chemical Industries Ltd. A paper on " The Use of Electricity in the Catering Industry " is being prepared by the Commercial Electrical Catering Apparatus Section of the British Electrical and Allied Manufacturers' Association and will be presented on their behalf by Mr. Felix A. Rogers, M.B.E.

The Convention Dinner and Dance, with accommodation for some 800 guests, will complete Thursday's proceedings.

On Friday, 1st July, two hours will be given to the popular " Electrical Forum," when questions are answered by a panel of experts, with the President acting as question master. This will be followed by the annual general meeting, at which officials for the next year are elected and the venue for the 1956 Convention is decided.

CENTENARY OF ALUMINIUM

On June 1st to the 11th, at the Royal Festival Hall, the aluminium industry is to present a Progress Exhibition, not only to mark the centenary of aluminium as a commercial metal but to demonstrate how the present vast output and unique experience already gained are now being applied to every great industry in increasing measure.

It was at the Paris Exposition in 1855 that the French scientist, Professor St. Claire Deville, who had succeeded in extracting aluminium from clay by a chemical process, exhibited samples of the metal. Aluminium was rated a precious metal, since it cost £60 per pound to produce.

The first article made of the new metal was a rattle for the child of the Emperor Napoleon III, who ordered aluminium cutlery to replace the gold plate at the Tuileries banquets. An aluminium watch-chain was also presented to the King of Siam, when on a State visit to France.

Attracted by its light weight, the Emperor commissioned Deville to produce aluminium on a large scale so that, with its cost reduced, it could be used for equipment for his soldiers. However, only 10 tons of aluminium were produced in the whole world in the next 10 years, its lowest price being £3 per lb.

In 1886, two scientists, Charles Martin Hall in America and Paul Heroult in France, working independently, simultaneously patented the electrolytic process for extracting aluminium from its oxide. Today using basically that same method of extraction, 30,000 tons of pure aluminium are produced in Britain alone, at the great hydro-electric plant in the Scottish Highlands, and world production in 1934 has probably exceeded 3 million tons and is still increasing. From £144,400 per ton its price is now £165 per ton.

A German scientist, Alfred Wilm, discovered in 1907, that by alloying small quantities of other metals with aluminium, its strength could be greatly increased. Further research and development in this country produced a wide range of alloys which made possible the development of the aircraft industry and of the internal combustion engine. Every alloy has been developed to exacting standards of uniformity and reliability.

The ten years that have followed the last war have been years of remarkable progress when research results have been applied to problems in every industry. First came the aluminium prefabricated house, 70,000 of which were built; to be followed by the adoption of aluminium in permanent building construction. In shipbuilding its success was never more strikingly demonstrated

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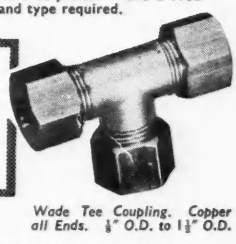

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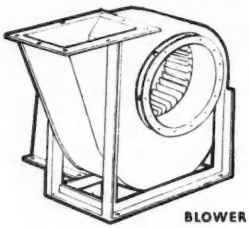

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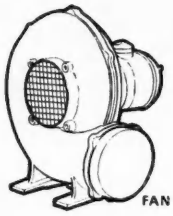

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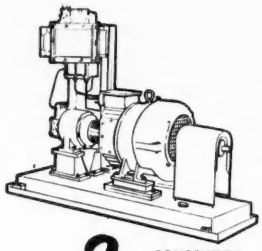

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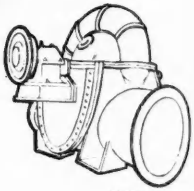

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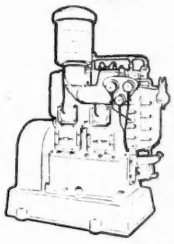

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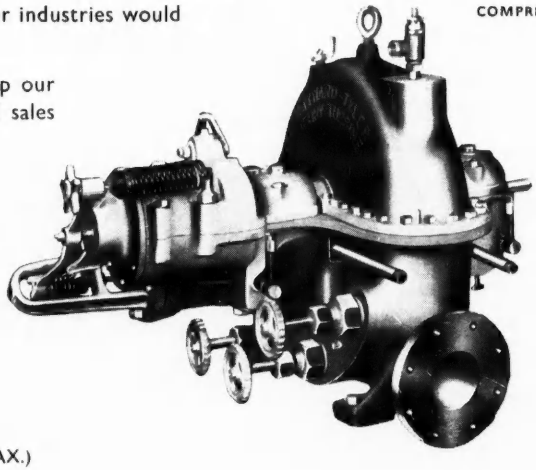

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than by the great American liner " United States," which has over 2,000 tons of aluminium in her superstructure, lifeboats, and many fitments. The world's first aluminium bridges have been built in Britain and aluminium coaches of London's Underground trains, and aluminium bus bodies on London's streets, provide ample evidence of the faith of the engineer in aluminium.

Its light weight and its resistance to corrosion is symbolised by Gilbert's famous aluminium statue of Eros, now more than 60 years old.

The Exhibition is primarily directed to all present and potential users of aluminium, particularly to professional engineers, designers and architects, representatives of Government, Local Authorities, and the great industries of Britain; to Commonwealth and foreign guests; also to teachers engaged in University and Higher Technological Education.

The Exhibition is being designed by Mr. Ronald Dickens, M.B.E., and will be divided into five sections:—Historical, Research, Production, Present Day Use of Aluminium and The Future of Aluminium.

FIRTH BROWN TOOLS LTD. RESEARCH LABORATORIES

The new Research Laboratories of Firth Brown Tools Ltd., which were opened in Sheffield by the Master Cutler (Mr. W. G. Ibberson) on January 13, 1955, are concerned with all aspects of tool manufacture, from control analyses on raw materials to the final testing of the cutting capabilities of the finished product. New methods of machining are kept under review and are the subject of special investigations, while a constant interest is kept in the design of novel cutting tools or new cutting materials.

The facilities required for the diverse functions of the department are mainly centred in the New Research Laboratories in Windsor Street. Certain sections however, are sited at strategic points in the works, where close liaison with the production side is necessary. These sections are: (i) Test Lathe for high-speed cutting with hard metal tools. (ii) Test machine shop, for the testing of drills, taps, saws, etc. (iii) Standards room.

The Research Laboratories comprise metallographical and analytical laboratories. The metallographical laboratory is equipped for the examination of the microstructure of tool steels and hard metals. The main equipment consists of a Vickers projection microscope for visual examination and for the preparation of photomicrographs. The usual apparatus for preparation of specimens is installed, including that necessary for grinding, lapping, and polishing with diamonds. Apparatus is also installed for " heat-tinting " specimens, a process the laboratory has developed for the microexamination of hard metals of the multi-carbide type, and which replaces the practice of etching. A darkroom for photographic work is provided adjacent to this laboratory.

The analytical laboratories comprise fully equipped spectrographical, chemical, mechanical-testing, heat-treatment, physical, and pyrometric laboratories. In addition, there is an experimental machine shop, in which test specimens are prepared. This machine shop is also equipped for the development of the new electrical methods of machining by the quenched-arc or spark-discharge processes.

THE COAL INDUSTRY IN EUROPE

The Organisation for European Economic Co-operation has just published the Report of its Coal Committee, entitled " The Coal Industry in Europe," in the form of a comprehensive survey of the coal situation in Western Europe during 1953 and 1954. In the light of the more recent trends observed, the Committee has endeavoured to make some suggestions as to probable developments in the coal industry in 1955. The Report analyses the various factors which influence the market, i.e. production, output, consumption, stocks, prices and trade. The general conclusions reveal a tempered optimism with regard to the short-term trend of the coal market in O.E.E.C. Member countries.

REVOLVING REFRACTORY CHART

The ability to substitute large monolithic units for the customary brick by brick construction, and the ease by which intricate special shapes can be quickly made or awkward corners filled up, has made the use of cold-setting refractories almost universal practice for a variety of jobs.

Some users make up their own refractory concrete from mixtures of aluminous cement and crushed well-graded firebricks or grog, while others prefer to use one of the proprietary cold-setting premixes.

Just as no particular make or type of firebrick is suitable for all conditions so does the type of refractory concrete, or even its suitability, vary, according to the particular conditions prevailing. To help those firms making their own mixtures to be certain that the mix that they are using is the best possible one in each circumstance, the Lafarge Aluminous Cement Co. Ltd. (manufacturers of Ciment Fondu aluminous cement) have produced a revolving refractory chart which shows at a glance the various uses for refractory concrete, the most suitable aggregates, suggested mixes and also the hot face temperatures which such mixtures should withstand. Copies may be obtained from the Lafarge Aluminous Cement Co. Ltd., 73 Brook Street London, W.1 free of charge.

EXHIBITIONS

BRITISH INSTRUMENT INDUSTRIES' EXHIBITION

The third British Instrument Industries' Exhibition to be held at Earls Court, London, from 28th June to 9th July 1955 will be the largest of its kind. It will show latest scientific and electrical instruments, industrial meters, automatic controls, scientific glassware, drawing office equipment, and allied products.

The organisers, F. W. Bridges and Sons Ltd., have made arrangements to cater for the needs of overseas visitors.

The exhibition is sponsored and actively supported by The British Electrical & Allied Manufacturers' Association, The British Industrial Measuring & Control Apparatus Manufacturers' Association, The British Lampblown Scientific Glassware Manufacturers' Association, The Drawing Office Material Manufacturers' & Dealers' Association and The Scientific Instrument Manufacturers' Association of Gt. Britain.

Concurrently with this Exhibition there is the 10th International Printing Machinery & Allied Trades' Exhibition at Olympia. Visitors to this will have the advantage of studying both modern printing machinery and the appropriate field of instrumentation, including temperature control.

New Uses of Isotopes are to be demonstrated from February 14th to 26th at the Birmingham Exchange and Engineering Centre.

Experts from the Isotope Division of the Atomic Energy Research Establishment at Harwell have devised the exhibition, and a number of industrial firms who manufacture equipment for work with radioactive isotopes will also be taking part.

Britain is now the largest supplier in the world of these radioactive materials, and the exhibition, which will be open daily from 9 a.m. to 5 p.m. including Saturdays, is mainly concerned with the many ways in which they can be used for industrial purposes and with the important part they are playing in the fight against cancer and other diseases.

Among the techniques demonstrated will be the use of isotopes as substitutes for X-rays in the examination of castings, welds and forgings for their soundness and freedom from blowholes and cracks; thickness gauges which can maintain a continuous measurement of thickness of metal sheet and plate, linoleum, paper, etc., and the thickness of plating, paint or lacquer coatings applied to metal surfaces; the measurement of leakage from water pipes and fuel lines—a method which has contributed to the reduction of

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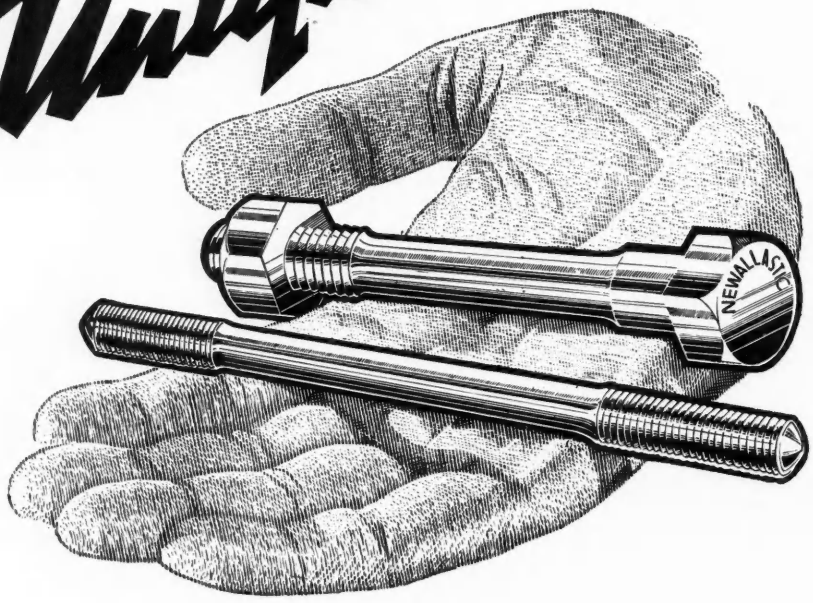

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fire hazard in aircraft; and improved methods of checking the efficiency of liquid or gas filters.

An interesting working demonstration will show how wear on bearings or moving parts can be measured in a few hours without having to dismantle an engine.

German Plastics Fair. October 8th to 16th in Düsseldorf, Germany. Organisers: Nordwestdeutsche Ausstellungs G.m.b.H., Ehrenhof 4, Dusseldorf. Agents: John E. Buck & Co., 47 Brewer Street, London, W.1.

Building Exhibition. November 16th to 30th at Olympia, London. Organisers: The Building Trades Exhibition, 4 Vernon Place, London, W.C.1.

The Second International Automation Exposition will be held at the Chicago Navy Pier from November 14th to 17th, 1955. Information can be obtained from the organizers, Richard Rimbach Associates, 845 Ridge Avenue, Pittsburg 12, Pa., U.S.A., or from The Exhibitions and Fairs Branch of the Board of Trade, Lacon House, Theobalds Road, London, W.C.1.

BUSINESS NOTES

Metal Industries Ltd., Victoria Station House, London, S.W.1, have formed a new subsidiary company, Metind Ltd., which will operate from a factory to be constructed in the Bromborough area. Mr. K. N. Swash has been appointed chairman and managing director of the new company and Mr. F. R. Livock and Mr. T. D. Turner have been appointed to the Board. Metal Industries Ltd. have also acquired the whole issued share capital of Finney Presses Ltd., of Berkley Street, Birmingham, whose activities will be co-ordinated with those of Fawcett Preston & Co. Ltd., another member of the Metal Industries Group.

The English Electric Co. Ltd. have recently taken over premises from the Ministry of Supply, at Whetstone, Leicester, which were previously occupied by the National Gas Turbine Establishment. The work to be carried out there by the English Electric Company will be largely research and development associated with power station engineering.

Industrial gas turbine development will continue, and in addition, extensive facilities for mechanical engineering laboratory work will be provided at Whetstone. Development work already being undertaken at the Company's Rugby works on steam turbines, water turbines, and diesel engines will be extended at the new premises.

British Insulated Callender's Cables Ltd. announce that the new address of their London Branch sales office is 10/14 White Lion Street, London, N.1. Tel. No. TERminus 8696 and 0372. The central administrative offices remain at 21 Bloomsbury Street, London, W.C.1.

The Morse Chain Co. Ltd. has changed the name of the company to Borg-Warner Ltd.

Mr. E. J. Burford, correspondent in the United Kingdom and Eire for Kovo Ltd., Motokov Ltd., Strojexport and Strojimport, of Prague, Czechoslovakia, announce that his address is now 30 Sloane Street, London, S.W.1. Tel. No. SLOane 0091.

Compression Ignition Ltd. has changed the name of the company to Twiflex Couplings Ltd.

Sheepbridge Engineering Ltd. announce that, by arrangement with Mahle-K.G., they have secured the sole U.K. manufacturing and selling rights for chrome plating light alloy engine cylinders and similar components by a special process. British van der Horst Ltd., a subsidiary of Sheepbridge Engineering Ltd., will operate this process by which it is now possible to deposit hard chromium directly on to aluminium alloys. The selling rights secured under the agreement will be administered by Sheepbridge Stokes Ltd.

Firth Cleveland Group Expansion. In addition to the recent acquisition of Keeton Sons & Co. Ltd., Sheffield, specialists in hollow bored shafts, and the production of sheet metal machine tools, Firth Cleveland has acquired Landmaster Ltd., Hucknall, Nottingham, manufacturers of horticultural rotary hoes and ancillary equipment. Landmaster Ltd. have recently purchased manufacturing and selling rights of the Colwood motor hoe, previously produced by Dashwood Engineering Ltd.

Other companies in the Firth Cleveland Group include Simmonds Aerocessories Ltd., The Firth Company Ltd., British Lead Mills Ltd., Stenor Ltd. and Firth Cleveland (Machine Tools) Ltd.

Firma Wilhelm Rump, of Gertrudenstrasse 11, Hamburg 1, have been appointed the shipyard representatives of Associated British Oil Engines (Export), Ltd., and National Oil Engines (Export), Ltd., subsidiary companies of The Brush Group, Ltd.

Glass Technology Scholarships. Students about to leave school, who have been studying Chemistry and Physics, should be attracted by Scholarships in Glass Technology at the University of Sheffield.

The Worshipful Company of Glass-Sellers of London offer up to four Scholarships which, in suitable circumstances, qualify for supplementation by the Ministry of Education up to the full value of State Scholarships. The Scholarship Examination consists of papers in Chemistry, Physics and Pure Mathematics (two as primary subjects and one as a secondary subject) together with an English Language paper.

Applications for entry should be made to the University by 26th February, 1955, but in subsequent years by the middle of January. Entry forms, syllabuses, etc., may be obtained from The Registrar, University of Sheffield, Sheffield, 10.

Entries for 1955 should be made immediately and the examination begins at the University on 21st March.

The British Glass Industry, with its centres in London, the Midlands, Yorkshire, Tyneside, Lancashire and Scotland, offers well paid posts to qualified men and the number of such posts is likely to increase for many years.

The Mond Nickel Fellowships Committee now invites applications for Fellowships of an approximate value of £900 to £1,200 for 1955. Fellowships will be awarded to selected candidates of British nationality with degree or equivalent qualifications to enable them to obtain wider experience and additional training in industrial establishments, at home or abroad, to make them more suitable for future employment in senior technical and administrative positions in British Metallurgical Industries. Each Fellowship will cover one full working year. Applicants will be required to state details of the programme they wish to carry out. Particulars and forms of application are available from The Secretary, Mond Nickel Fellowships Committee, 4 Grosvenor Gardens, London, S.W.1. Completed application forms are required by 1st June, 1955.

The Council of The Institution of Mechanical Engineers have awarded the 1955 James Watt International Medal to Dr. Igor Ivan Sikorsky, of the United States, in recognition of his life's work in applying science to the progress of mechanical engineering through pioneering development of aircraft of various types: multi-engined, amphibians, and particularly helicopters.

The James Watt International Medal was founded to commemorate the bi-centenary of the birth of James Watt, which took place on 19th January 1736.

BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 2 Park Street, London, W.1)

Flanged Steel plug valves for the Petroleum Industry (excluding well-head and flow-line valves). B.S. 1570 : 1955 specifies requirements for flanged steel lubricated and non-lubricated plug valves having straight-way ports. Three-way and four-way valves are not covered by this standard. The range of sizes in most classes is from 1 to 24 in. with primary service pressure ratings of 150 to 2500 lb/sq. in. It deals with design and manufacture,

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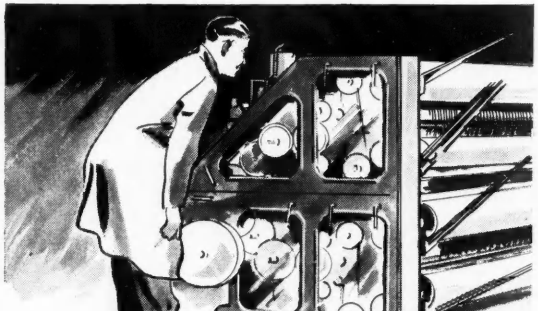

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materials, workmanship, tests, marking, inspection and despatch. Wherever possible, this standard is intended also to apply to valves fitted with internal body or port liners or applied hard facings on either bodies or plugs. Price 10/-.

Spring Collets. B.S. 2553 : 1954 should have considerable impact on workshop economy in the future.

A committee charged with the preparation of the standard designed with the active co-operation of some machine tool designers, a range of collets in nine main types which, while being different in several minor respects from all existing collets, could be produced by all manufacturers with the absolute minimum of dislocation. The only prerequisite to this was that the designs for the machine noses should in future take full account of the dimensions and tolerances of these collets. Price 5/-.

Cable Sealing Boxes for Oil-immersed Transformers. Part 1: Boxes for Voltages up to and including 11 kV. B.S. 2562. Part 1 : 1955 deals with a range of cable sealing boxes of the type in which the cores of the cables end, and are connected to terminals, within the box itself.

The standard was prepared primarily for lead covered cables and account has been taken of the use of cables with an aluminium sheath. The boxes are classified according to voltage, current rating, number of poles and the number and size of cables and cable glands. Standard dimensions are given for a range of boxes which are for use indoors or outdoors and also for certain components of the boxes.

In addition to seven tables, eighteen figures are included which illustrate the method of classification. The figures also give standardized dimensions of the boxes in each class.

Although the standard does not purport to specify complete details of design and construction, some observations of this aspect are included for guidance.

It is intended to issue in due course a second part covering boxes for voltages in excess of 11 kV. Price 7/6.

CONTRACTS

Aluminium Wire & Cable Co. Ltd. have been awarded a contract for over 250 miles of large sized steel-cored aluminium conductor to an approximate value of £114,000 sterling for the new 220 kV electrical grid system in Tasmania.

Brush Electrical Engineering Co. Ltd., Loughborough, has received an order for electric motors, totalling 4,800 hp, from Vaal Reefs Exploration and Mining Co., Ltd., South Africa.

The motors are of the high torque, squirrel cage, induction type, in 27 sizes ranging from 60 to 250 hp. They will be used for gold mining, driving crushers, and similar works.

The order was placed through Brush Aboe (Southern Africa) Ltd., the Brush Group's subsidiary company in Johannesburg.

The General Electric Company Limited announces that it has received an order from the Johannesburg Municipality to supply for the Kelvin Power Station three 30 MW turbo-generator sets complete with feed heating and evaporating plant, bringing the total number of sets on order or under construction for this station to six. The value of this latest order is nearly £1 million.

The turbines will operate under steam conditions of 600 p.s.i.g. at the turbine stop valve and a temperature of 850° F. The machines are of two casing design with 22 stages of high pressure and 6 stages double flow of low pressure. The 37,500 kVA alternators, which are air cooled, generate at 11 kV and have direct coupled exciters.

Each set is provided with a 4 stage feed heating plant giving a final feed temperature of 356° F at full load, designed and manufactured by Vickers-Armstrongs Ltd.

The consultants to the Municipality are Merz and McLellan Ltd., Newcastle.

Transformers (Watford) Ltd., Sandown-road, Watford, Hertfordshire, who recently announced that they had introduced, for the first time in this country, the manufacture of class-H air-cooled silicone-impregnated distribution transformers, now announce that they have been successful in securing a large contract, in Italy, for similar class-H transformers.

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The rate for all classified advertisements is 6d. per word ; in bold print 9d. per word ; minimum order 6s. Box-number advertisements 1s. extra. Instructions, together with remittance, must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

APPOINTMENTS VACANT

QUALIFIED SALES REPRESENTATIVE required to sell technical products. Age 27-35. Good personality essential. Sound engineering background and practical experience in either the structural, marine, or aeronautical fields desirable. Write in confidence, stating professional qualifications, to Box No. 24.

SENIOR ENGINEER.—An international light engineering Co., in West London, has a vacancy for a senior engineer to develop specialised processes for the manufacture of magnetic recording heads. Applicants must have development and production experience on magnetic recording equipment and be qualified to exploit new materials and methods of manufacture. Preference will be given to applicants with metallurgy qualifications. The post is pensionable and the salary will be commensurate with ability. Apply Box. No. 25.

MACHINE TOOL REPRESENTATIVE required for London Territory. Those with experience in production engineering are invited to apply, giving details of experience in chronological order, age and salary expected, to The Secretary, **Charles Churchill & Co. Ltd.,** Coventry Road, South Yardley, Birmingham, 25.

ROCKET DIVISION.—Armstrong Siddeley Motors require Designers and Draughtsmen for exacting work in this division. Applicants must possess H.N.C. and/or have considerable drawing office experience. Apply, quoting Reference CG/RD5, Technical Personnel Manager, **Armstrong Siddeley Motors,** Coventry.

SENIOR PROJECT ENGINEER.—A vacancy is available for a Senior Project Engineer to head a group for the production of power from atomic energy. Applicants should have a background of experience, preferably in marine engineering, or, alternatively, experience on the mechanical side of the generation of power for land purposes. In addition, applicants should have scientific qualifications in the form of First-Class Honours Degree in Marine or Mechanical Engineering, together with a sound practical experience. This is an extremely important position and a good salary will be paid to a man meeting the stated requirements. Employment will be at Harwell initially. Box No. 26.

Armstrong Siddeley Motors, Coventry, have vacancies for

TECHNICAL WRITERS
PLANNING ENGINEERS

for work on gas turbine aero engines. Apply to Reference D1, Personnel Manager, **Armstrong Siddeley Motors,** Coventry.

TEST BED ENGINEER required by Midland Motor Car Manufacturer for interesting engine development programme. Apply to Reference FWA.1 stating theoretical and practical experience. Box No. 27.

MECHANICAL ENGINEER required for work on the design of small electro-mechanical equipments. Applicants should have served an engineering apprenticeship, or the equivalent, together with several years' experience in the field of design. Small model shop facilities available to Engineer. Greenford area. Pensionable and salary commensurate with ability. Apply giving age and fullest details of experience to Box 20.

QUALIFIED ELECTRICAL ENGINEER required for interesting new development of established process in Central London Laboratory. Primary considerations are sound knowledge of fundamental principles and aptitude for their practical application. Pension Scheme. Box No. 29.

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WANTED

MARCH, 1953 ISSUE of "The Engineers Digest" wanted at 4/6 per copy. Subscription Dept., 120 Wigmore Street, London, W.1.

BOOKS RECEIVED

The Practical Engineer Pocket Book. 65th edition, 1954. Edited by N. P. W. Moore, B.Sc., A.C.G.I., D.I.C., A.M.Inst.F. 789 pages. Publishers: Sir Isaac Pitman & Sons, Ltd., Parker Street, Kingsway, London, W.C.2. Price: 12/6.

As always, this handy little volume contains a wealth of practical information for engineers. In the preparation of this new edition, further revision has taken place. Thus, for instance, an entirely new section has been added on the subject of Industrial Hygiene, the section on Bearings and Mechanical Transmission has been largely rewritten, and the section on Locomotives has been revised and extended.

In the preparation of new material and in the revision of existing sections, an endeavour is made to present sufficient of the underlying principles to permit the reader to extend the examples and data given to his immediate probelms. The practice of giving clear references to more complete sources of specialised information has been retained and extended. In addition, there are useful technical dictionaries in German, French, and Spanish.

Electrical Trades Directory, 1955. 73rd edition. Edited by Norman H. Codling, D.F.H.(Hons.), A.M.I.E.E. 1615 pages. Publishers: Benn Brothers Limited, 154 Fleet Street, London, E.C.4. Price: 50/-.

This new edition, with more than 65,000 references, is a comprehensive and reliable store of information concerning the electrical and allied industries and trades, the products they manufacture, and the services they offer. These details, which are not to be found in any other single publication, are classified in appropriate sections, with an index for easy reference. This year, the list of manufacturers has been increased by more than 260 names, bringing the total in this section to over 6600. The numbers of engineers, electrical installation contractors, electrical and radio wholesalers, and distributors also show increases. Similarly, in the Products and Materials Section, the number of headings has been increased to 2180, and the references in this part of the Directory now exceed 45,000. The Trade Names Section—unique in its extent—now contains more than 5300 names of patented products.

Kempe's Engineers Year-Book. 60th edition, 1955. Edited by C. E. Prockter, M.I.E.E., M.I.Mech.E., under the direction of B. W. Pendred, M.I.Mech.E., M.I.S.I. 2810 pages, approx. 1500 illustrations. Publishers: Morgan Brothers (Publishers) Ltd., 28 Essex Street, Strand, London, W.C.2. Price (two volumes in case) 75/- (postage 2/6).

As will be remembered, the 1954 edition of this well known reference book was completely rewritten and the whole work reset in a clearer typeface. In this new edition for 1955, attention has been primarily, though not wholly, confined to detail. Thus, for instance, additional information is included on the subject of shell moulding in the chapter on Foundry Practice; on geared couplings in the chapter on Shafts and Couplings; and on preparing rolls in the chapter on Forging Hammers and Drop-Forging Plant. Furthermore, certain additions have been made to the chapters on Electronic Engineering, on Explosives, and on Flow Metering and Mechanical Testing. In this edition, too, the number of cross-references in the text has been increased.

AEG Manual. 703 pages, 1063 illustrations. Published by Allgemeine Elektricitäts Gesellschaft, through Verlag W. Girardet, Essen, Germany, and distributed in U.K. by Welmec Corporation Ltd., 147 Strand, London, W.C.2. Price: 42/-.

This authoritative and excellently produced English edition of the AEG Manual is based on the seventh edition of the "AEG Hifsbuch für elektrische Licht- und Kraftanlagen." Though originally published for the benefit of newcomers to the field of electrical engineering in general, and to electrical installation practice in particular, it has been brought into line with the latest international developments and is undoubtedly a practical and useful reference work for most electrical engineering applications.

Quite apart from the concise and clearly expressed information in the text and a wealth of accompanying illustrations, the manual merits special praise for its comprehensive index, its symbols and diagrams of connections, and its collection of conversion tables, covering a multitude of international engineering units. In short, a most useful book, well conceived and produced, and, in its field, comprehensive.

Irrigation and Hydraulic Design (Vol. I). By Serge Leliavsky, Ph.D., M.I.C.E., M.Am.Soc.C.E. 505 pp., 314 illustrations. Publishers: Chapman & Hall Ltd., 37 Essex Street, London, W.C.2. Price: 126/-.

This book may well become an indispensable work of reference for the irrigation and hydraulic designer in his daily work. Its scope is comprehensive and it contains numerous formulae, methods, and design theories which have been collected, replenished, and kept up to date during the forty years technical life of the author, who is well known as an authority on his subject.

The book has been planned in three volumes. This first volume, entitled "General Principles of Hydraulic Design," contains the more general fundamentals of design work, in particular concerning percolation in the granular foundations beneath hydraulic structures and in earthen banks, tail erosion in its various aspects, and hydraulic flow in canals and rivers, both uniform and non-uniform. The book is profusely illustrated with diagrams, charts, and photographs.

Vols. II and III are in course of preparation and are entitled "Irrigation Works" and "Hydraulic Structures for Irrigation and other Purposes," respectively.

LATEST INDUSTRIAL LITERATURE

1. Industrial Stoving Equipment. Useful technical information, together with numerous photographs of typical examples of industrial stoving equipment, is contained in a well produced and illustrated brochure of 20 pages.

The wide range of ovens described and illustrated includes both stationary and conveyor ovens, with heat transfer by natural convection, forced convection, or radiation. Standard designs and sizes of ovens can be supplied, but, where these are not suitable for any specific requirement, redesigning can be effected to suit customers' needs.

2. Elastic Drive and Cotter Pins. Two leaflets describe and illustrate a range of elastic pins intended to reduce very considerably labour and machining costs, particularly where their application replaces a component which calls for a reaming or similar machining operation. The only operations required in the use of these pins are the drilling of the hole and pressing in of the pin. The resulting assembly is of adequate mechanical strength and is safer in cases where vibration is a problem. Furthermore, they can be removed and inserted a number of times without damage.

A supplementary folder describes a range of retaining rings for spindles and shafts, with many outstanding advantages.

3. Pneumatic Remote Control. Details are given in a 4-page brochure of a system which provides accurate remote control for the operation of valves to any position, from fully open to closed, but still leaves the valve free for immediate manual operation, when required. The air motor used is of the diaphragm type, and the speed of rotation varies with the load. No maintenance is necessary, apart from periodic refilling of the oil-mist lubricator.

The pneumatic operation of the system ensures a flameproof unit and provides extreme simplicity of installation. These units are suitable for incorporation in automatic, programme, and other control systems.

4. Electric Lamps. In a well conceived and executed 60-page catalogue, a comprehensive range of electric lamps is described and illustrated. More than 1500 of the more important lamps in the range are listed in groups and indexed under convenient sections, each of which consists of two facing pages, with full details for easy reference. Sections are also included dealing briefly with lighting equipment and fittings, and a ready reckoner.

The range includes a new series of fluorescent lamps, a 400-W addition to a range of colour-corrected mercury discharge lamps, a new 1-kW horizontally operated mercury lamp, and a new 10-kW projector lamp for film studios, the largest tungsten lamp made in Great Britain.

5. Cast-Iron Conveyor Pulleys. In a neatly produced, illustrated 4-page brochure, details are given of a range of cast-iron conveyor pulleys for medium and heavy duties. These pulleys are available in single and double arm or plate centre construction, dependent on belt width, and as taper (convex)-faced or flat-faced pulleys. The standard range of diameters is from 12 to 60 in., with face widths varying from 16 to 64 in. Pulleys to suit customers' requirements can, however, be made at short notice.

Rubber-lagged pulleys can be supplied in cases where the materials handled have an adverse effect on the normal driving efficiency of the pulley. Furthermore, fabricated steel conveyors pulleys are also available for light, medium, and heavy duties.

6. Ball and Roller Bearings. The care needed in the storage, mounting, lubrication, protection, and dismantling of ball and roller bearings is not always fully understood. A neat and very useful folder, designed for easy reference, helps to get the best out of bearings by giving practical hints on their maintenance, etc.

7. Cutting-Speed Indicator. A leaflet describes and illustrates an indicator designed to measure the cutting speeds used in measuring components, and to ensure that the selected cutting speed in turning, boring, and planning is within correct limits. The information is shown directly in feet per minute, and the instrument can therefore be also used for determining any surface speed.

A noteworthy feature of the indicator is its operational simplicity, coupled with its positive indication, enabling it to be used for checking optimum machining speeds by unskilled labour. Nevertheless, it is also useful for planning and production engineers, for making any desired speed adjustments on the shop floor.

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● *WATER-COOLED ELECTRIC MOTORS*

Totally enclosed electric a.c. motors are usually provided with a cooling arrangement which uses air as a coolant; where large motors are concerned, cooling-air requirements are considerable. This has led to the use of circulating systems in which the heated air is passed through a water cooler and then returned to the motor in a closed circuit. A type of motor has recently been marketed, in which the cooling water is passed directly through both the motor stator and rotor, in parallel or series flow. It is claimed that this method of motor cooling is far more efficient than any other kind of cooling as applied to totally enclosed motors, and that water-cooled motors are about as efficiently cooled as motors with direct ventilation. Other advantages of the new type are said to be low unit weight and small space requirements. Since the cooling effect is independent of the ambient air temperature, these motors can be installed in very hot locations. In order to avoid scale deposition in the cooling passages, it is stated that the cooling-water inlet temperature should not exceed 20 to 25 C. Cooling-water requirements are reported to be about 22 gallons per hour per kW to be dissipated. A motor of 1000 kW nominal rating at 1500 rpm will therefore require some 800 gal of water per hr. A thermostatic control is incorporated which shuts the motor down if the water supply fails. It is suggested that water cooling could also be applied in cases where space considerations require the use of exceptionally highly rated small motors.

● *SOLUTION-TYPE CERAMIC COATINGS*

A novel type of refractory oxide coating recently developed should eventually find widespread application for a variety of purposes. These coatings are easily applied, since the base material of the patented composition is a true water solution rather than a suspension of solids in water. The film is formed by spraying the solution onto the surface to be coated. The liquid medium is a solution of a metallic salt which, by spraying onto a sufficiently heated surface, is reduced to the oxide. Where a refractory coating is to be formed, zirconium and chromium salts are used. Decomposition to the oxide leaves the very small particles firmly adhering to the surface, the bonding apparently being produced by molecular attraction. The temperature at which decomposition occurs and to which the surface must be heated during spraying varies from 400 to 450 F in the case of ammonium zirconyl carbonate, which is decomposed to zirconia, while for the decomposition of chromium nitrate to chromium oxide a temperature of about 700 F is required. The coatings produced are porous and composed of numerous extremely fine crystallites. The coatings can be built up to a thickness of about 0·01 in.; they are not particularly hard, and can be scratched with a knife blade. Coated materials can be moderately bent, twisted, or elongated without damage to the coating but, if sharply bent, the coating comes off as a powder. The films protect the base material against corrosion by solids or viscous liquids, particularly at elevated temperatures and against atmospheric attack at high temperatures. The dielectric strength of the films seems to offer another field of application. Zirconia films on mild steel have shown a d.c. resistivity of more than one megohm per cm^3 at room temperature.

● *MEASUREMENT OF GAS-TURBINE BLADE TEMPERATURES*

A greatly improved system has been evolved for measuring the temperature of gas-turbine blading during operation. This system includes special high-temperature resistance thermometers capable of withstanding large centrifugal forces, an inductive commutator for transmitting the signal from the turbine rotor to a stationary receiver, and electronic equipment for converting the signals into temperature readings. It is reported that, with the new equipment, temperatures can be ascertained with an accuracy of ± 25 F at temperatures as high as 1500 F, regardless of the rotor speed. This system, which supersedes the usual arrangement of blade-mounted thermocouples connected to brushes or slip-rings, relies on the use of an inductive commutator consisting of a number of pairs of rotating and stationary induction coils. Rotation of the rotor brings each pair of revolving coils into and out of coupling with a pair of stationary coils. One stationary coil is excited with current at a low radio frequency, while the other stationary coil serves as the source of output voltage. On the rotor the coil-pairs are electrically connected together and shunted by the variable-resistance thermometer element. One of the

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rotating pair couples electromagnetically with the exciting stationary coil, and the other with the output coil. The result is that the amount of energy transferred from the rotating to the stationary system is governed by the resistance of the thermometer element when its particular rotating coil pair is coupled to the stationary coils. With optimum design, the carrier-frequency voltage output is very nearly a linear function of the thermometer resistance. The magnitude of the output is independent of the speed of rotation. A special type of resistance thermometer of small dimensions has been developed, the external diameter being 0·04 in.

● *SYNTHETIC POLYESTER FELT*

A synthetic heat-resistant felt has been developed, consisting of polyester fibres which are combined to produce a structure similar to that of wool felt. It is claimed that the new material has three times the breaking strength of wool felt and twice its splitting resistance, and after exposure for two days in moist air at 350° F retains its strength sixteen times better than wool felt. The synthetic felt is said to be non-swelling, to be unaffected by mildew and micro-organisms, and to be easily cleanable. Good resistance to acids and alkalis, high deformation recovery, and good dimensional stability and uniformity are other advantages claimed for the new material. The polyester fibres have neither the interlocking tendency nor the surface which are the basis of conventional wool-felting techniques. A new felting process had therefore to be developed. This consists in running a batting of prestretched fibres, carded like cotton batting, through a needle loom, where barbed needles pick up fibres from one side of the batting and carry them through the other side, thus stitching the material together with its own fibres. The resulting material is then compacted by exposing it to dry heat at between 300 and 400° F, or by immersing it in boiling water. The fibres shrink by from 30 to 50% when exposed to heat, and actual shrinkage can be controlled to produce felts of any desired density. It is thus possible to determine the properties of the polyester felt by adjusting the degree of fibre shrinkage, the amount of needle punching, and the weight of the carded batting.

● *NEW TYPES OF SILICONE FOAMS*

Several types of high-temperature silicone foams can now be produced from pre-mixed powders containing the exact proportions of resin, filler, blowing agent, and catalyst. The foamed product is made by pouring a given amount of powder into a cavity or mould and heating it at a selected temperature, depending upon the desired density of foam. One of these new materials can be foamed to densities ranging from 10 to 14 lb/cu ft, another to densities ranging from 12 to 16 lb/cu ft, and a third to densities ranging from 14 to 18 lb/cu ft. The first-mentioned powder is the most resistant to thermal shock; samples have been cycled repeatedly between room temperature and 600° F without cracking. At 700° F, this material withstood a three-day exposure with only slight dimensional change, total weight loss being less than 10%. Although the foam has good compressive strength at room temperature, it loses most of its original strength at elevated temperatures and becomes soft and rubbery. This tendency can, however, be utilised for forming the foam to a desired shape before carrying out final post-curing. After the initial expansion, sections can be cut from the foam and reheated to temperatures around 200° F. At this temperature the foam softens and can be wrapped around such shapes as air ducts, pipes, and tanks. These silicone foams, when foamed in place, will adhere to most metals and glass-cloth laminates. The moulds do not require preheating, but the powder is poured into the cavity and heat is then applied by air-circulating ovens, strip heaters, heat lamps, or similar equipment. In many applications, no post-cure will be required, as the foam will cure and harden during use.

● *ELECTRONIC DEPTH GAUGE*

A precision-type electronic depth gauge, operating on entirely new principles, has been developed and is able to indicate the depth of a submarine to within 6 inches and to register accurately any rate of depth or climb. Alternatively, it can be used for measuring the height of liquids in a tank. Thus, for instance, it could register at any convenient location the quantity of oil in gallons in any tank in a tank farm to one part in 4000; in this case, an additional advantage of the device is that there is no need to take into account variations in oil density due to temperature. The heart of the instrument is a special hydraulic pressure cell, the output of which is amplified three million times by four stages of sub-miniature thermionic valves, further amplification being achieved by means of a magnetic amplifier. The output of the latter is fed to a servomotor, which drives the rotating member of a precision potentiometer forming part of a Wheatstone-bridge circuit. A change in pressure on the cell, i.e., a change in depth, causes the potentiometer to seek a new position to restore voltage balance. The potentiometer position, which can be shown either on a counter or a dial, is then an accurate indication of depth. Where it is necessary to show the rate of change of depth, a small alternating current generator is fitted on the shaft of the servomotor and the output voltage of the generator will then be proportional to speed, and hence will also be proportional to the rate of change in pressure. By using pressure cells with

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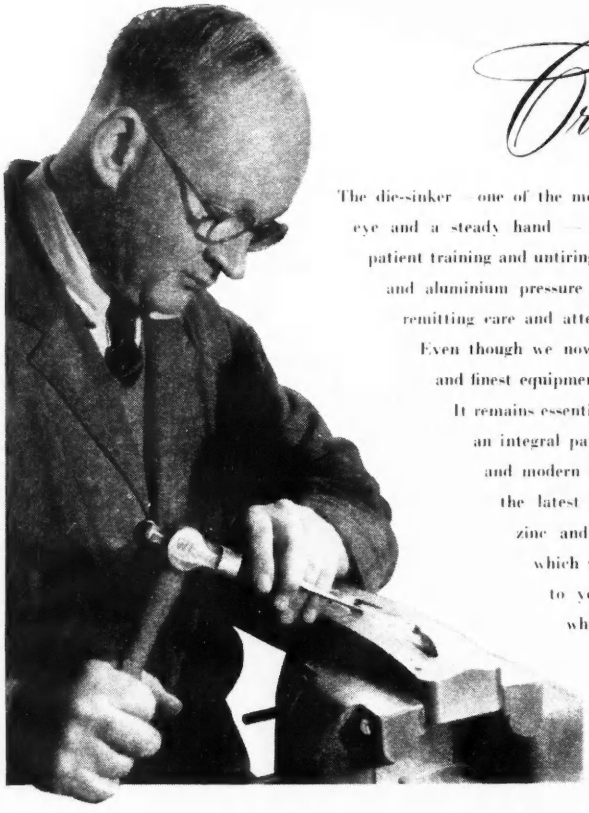

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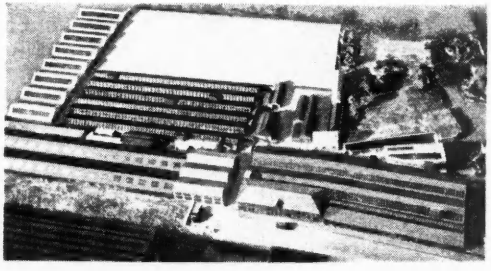

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different ranges the gauge can be made to measure pressures corresponding to depths ranging from a few feet to many thousand feet. Furthermore, by using strain-gauge-type load cells instead of a hydraulic pressure cell, the instrument can be used to show total weight and variations in weight.

● *RESILIENT COUPLING FOR SHIP-PROPULSION MACHINERY*

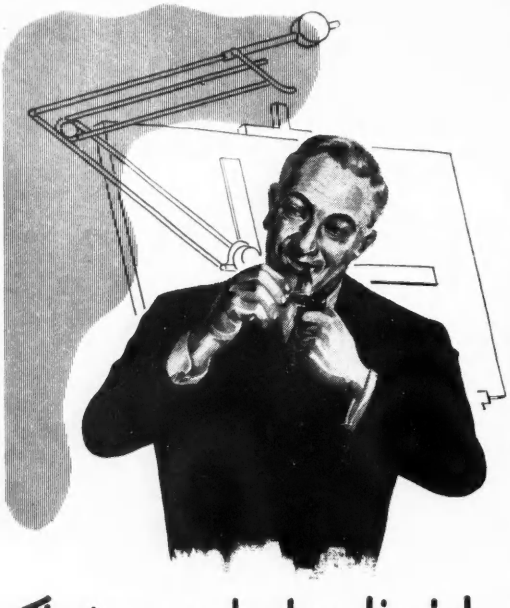
A new design of highly resilient coupling for ship-propulsion plant is claimed to have proved its merits in actual service and two types are now commercially available. One type is suitable for slow-running engines with speeds up to 375 rpm, while the other type has been developed for application to medium-speed engines running at 600 to 700 rpm. The low-speed coupling is of especially simple design, consisting of two coupling discs, a rubber sleeve, two pressure rings, and a number of screws. A characteristic feature of the coupling is the arc-shaped profile of the highly elastic rubber sleeve. This sleeve is manufactured from a rubber compound of 80 Shore hardness, and its torsional elasticity is so high that it permits an angular displacement between the coupling halves of 5 to 10 deg. at normal service torque. An important advantage of the coupling appears to be the fact that the rubber sleeve may be mounted or removed without requiring any axial displacement of any shafting. This is achieved by making the sleeve in two segments. Because of the centrifugal forces acting upon the sleeve, a peripheral speed of 100 fps must not be exceeded but, even so, the largest type available permits the tramsmission of 2000 shp at 375 rpm. In the medium-speed coupling, which is of somewhat different design, a rubber compound of 72 Shore hardness is used, and the rubber part is a simple cuff-shaped piece. In consequence, a peripheral speed of 200 fps is permissible. The largest coupling of this type can transmit up to 1000 shp at 700 rpm. Owing to the flexibility of the coupling, it may be used in shafts of main machinery, resiliently mounted on rubber to isolate vibrations from the ship's hull; actual installations of this type have already been made in small passenger vessels.

● *IMPROVED HYDRODYNAMIC BEARING*

In hydrodynamic bearings of conventional design, a layer of lubricant exists between the shaft surface and the bearing support, the space for the lubricant being usually of wedge-like form and bounded by arcuate surfaces. In a recently accepted patent application, it is pointed out that all such bearings, in which the lubricant space has a fixed and predetermined shape, have only a small supporting capacity, because manufacturing accuracy and hence minimum height of the lubricant space are limited. Moreover, rotation can take place only in one direction and this limitation also applies to bearings of the tilting-pad type, as well as to bearings in which the axle brass is thinner at the ends. The inventors claim that these disadvantages are obviated by their design, in which the bearing comprises an outer ring containing a resilient bearing part which, when at rest, is positioned to and conforms with the surface of the shaft. Load-distributing pieces are provided between the resilient member and the outer ring, and these pieces are so shaped that the pressure exerted by the load is at a maximum in the middle and zero at the ends of the resilient load-bearing part. A thin film of lubricant of substantially uniform thickness is formed between the rotating surface and that of the resilient bearing part, which is made of plastic material, usually 1 mm in thickness. It is advantageous to keep the lubricant pressure constant in the axial direction, so that there is no axial pressure gradient and therefore the lubricant does not flow towards the sides of the bearing. Owing to the elasticity of the plastic material of which the bearing member is made, the edges of the latter are in contact with the rotating surface and prevent outflow of lubricant at this point. Rotation of the shaft may be in either direction.

● *MANUFACTURE OF WIRE DRAWING DIES*

The formation of the drawing passages in diamond or hard metal dies for the drawing of fine wire requires exceptional precision and has hitherto been effected mechanically by grinding or drilling, with a high degree of wear on the tool. A recent invention envisages making the drawing passages by bombardment of the die material under sub-atmospheric pressure with electron or other corpuscular beams, in which the kinetic energy of the corpuscles is sufficient to vaporise the die material at the point of impingement. The kinetic energy of the corpuscles contained in the beam is almost entirely converted into heat at the point of impingement on the die material by the braking of the motion of the corpuscles. The magnitude of the kinetic energy is given by the product of the acceleration voltage and the current intensity of the beam. Since it is possible to use beams with an accelerating voltage of 100 kV and a current intensity of 10 mA, about one kW is available at the cross-section of the die passage, which may vary from 0·0001 to 1·0 mm^2. This energy is adequate for producing rapid vaporisation of the material. Moreover, since the intensity at the margin of the pencil of rays drops to zero in a small finite interval, a thin layer, in which the material no longer vaporises, but is only fused, will be formed in the marginal zone of the drawing passage. It is claimed that in this way a particularly smooth passage is formed, with a favourable influence on the surface formation of the drawn wire. By varying the angle of the pencil of rays during "drilling," it is possible to shape the drawing passage to any desired contour.



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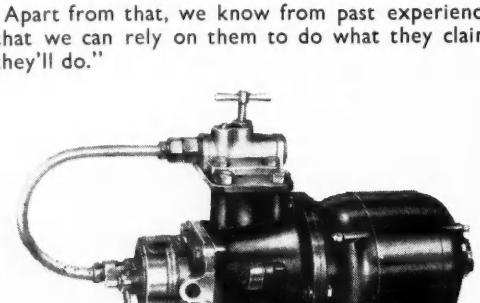

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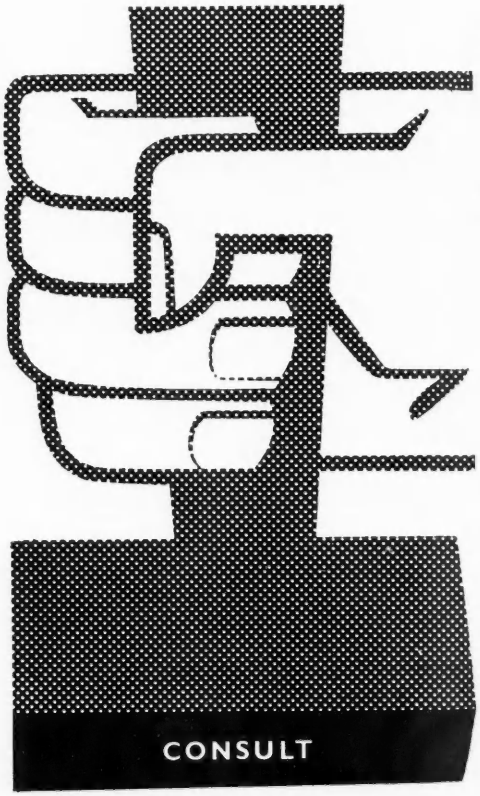

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● *MEASURING LUBRICANT THICKNESS ON POWDER-METALLURGY DIES*

The lubrication of dies used in the production of powder-metal compacts may be effected by the addition of lubricant to the powder charged into the die or by applying lubricant to the die walls. For the latter method, the thickness of the lubricant coating is critical and must be controlled carefully over the entire working area of the die. An accurate and practical method for measuring lubricant thickness is necessary to develop acceptable procedures for applying lubricant and for controlling these procedures during production. Considerable efforts have therefore been made towards evaluating various methods of measuring the film thickness of lubricants. In a recent investigation concerned with measuring the thickness of the lubricant in a round die, it was found that an air-gauge system provided the only satisfactory solution. This system employs a mandrel which is placed in the bore of the die, an air stream being passed from a flow meter into the mandrel, whence it flows out through two radial holes. After leaving the radial holes, the air stream passes between the outside diameter of the mandrel and the bore of the die. Since the air is supplied at constant pressure, the greater the clearance between mandrel and die, the greater will be the air velocity. By measuring bores of known diameter, the air-flow readings can be calibrated to read true bore diameters. The thickness of a coating on the die bore is determined by measuring one element line of the bore before and after being lubricated. The thickness between the two readings will then yield twice the average coating thickness. It is reported that the thickness of coatings may be determined to an accuracy of about $\pm$ 0·00005 in. over a range from 0·0001 in. to 0·001 in. thickness. The reproducibility of readings is said to be within 0·00005 in.

● *SENSITIVITY OF MECHANISMS TO TRANSIENT ACCELERATIONS*

The concepts of transient analysis, although frequently applied to electrical problems, are seldom used in the discussion of mechanical transients or "shocks." The result has been that, in describing the shock resistance of a structure, statements are often made to the effect that a structure will resist an acceleration of, say, 50 *g*, with complete lack of regard for the fact that time is a variable in a dynamic problem. Although those familiar with the problems of transient analysis are apt to be more cautious in their statements, there does exist a general state of confusion regarding the role of duration of acceleration and natural frequency of the structure undergoing acceleration. A recent paper discusses the factors which influence the response of the mechanism being subjected to shock and describes a method of presenting transient data and the resulting response. The author of the paper also suggests rational data for design for shock resistance or for the actuation of inertia mechanisms. Although the nature of the problem is such that exact design formulae may not be developed, the suggested approach to the problem is such as to make it possible to recommend the most probable design factors and to indicate the possible spread of performance under varied operational conditions. The particular problem investigated in this paper is an inertia-operated switch in which the switching member must move a prescribed distance against a restraining force in order to perform its switching action, in response to a transient forcing action. The conclusions arrived at for this mechanical system, with certain modifications and simplifications, can also be applied to the analysis of the behaviour of shock-resistant structures, since failure of a structure is characterised by elastic or plastic strain.

● *NEW ALLOY FOR CORROSION-RESISTANT STEEL SPRINGS*

A recent patent relates to springs and spring materials, especially watch springs, which must be resistant to corrosion and at the same time must possess very good spring properties, such as high modulus of elasticity, tensile strength, and fatigue strength at low as well as normal and high temperatures. The materials which have been used so far for this purpose are either very expensive, often containing alloying elements which influence cold and hot working adversely, or carbon steels with rust-preventive coatings. Such coatings do not, however, always offer safe protection against corrosion and they may be difficult to apply, in addition to being expensive; furthermore, they may cause difficulties such as hydrogen brittleness. For a long time it has been felt desirable to produce springs from the well-known austenitic stainless steels, particularly of the 18 8 type. However, such austenitic steels cannot be given improved mechanical properties by means of ordinary hardening methods, as they remain austenitic down to room temperature. On the other hand, a certain hardening can be produced by cold working. The invention referred to relates to an unstable austenitic steel which contains from 6 to 10% nickel, 12 to 20% chromium, and at least one additional ferrite-forming element, such as silicon and or molybdenum. A certain percentage of columbium, tantalum, vanadium, and tungsten may also be included. In addition, manganese, copper, and cobalt may also be included in prescribed percentages.

For further information on developments reviewed in THE BLUE PAGES, readers are invited to write to the Editor.

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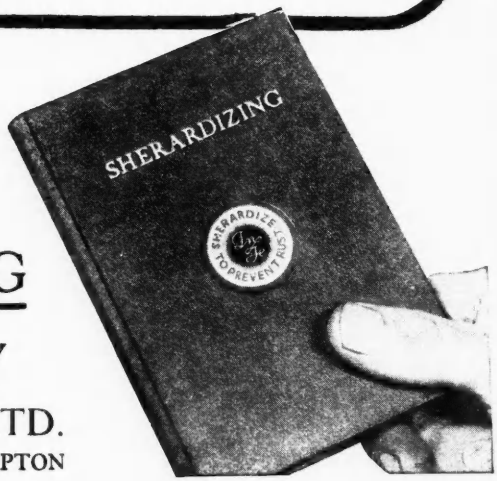

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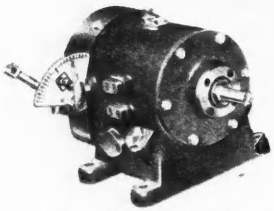

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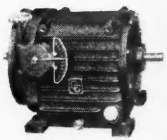

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